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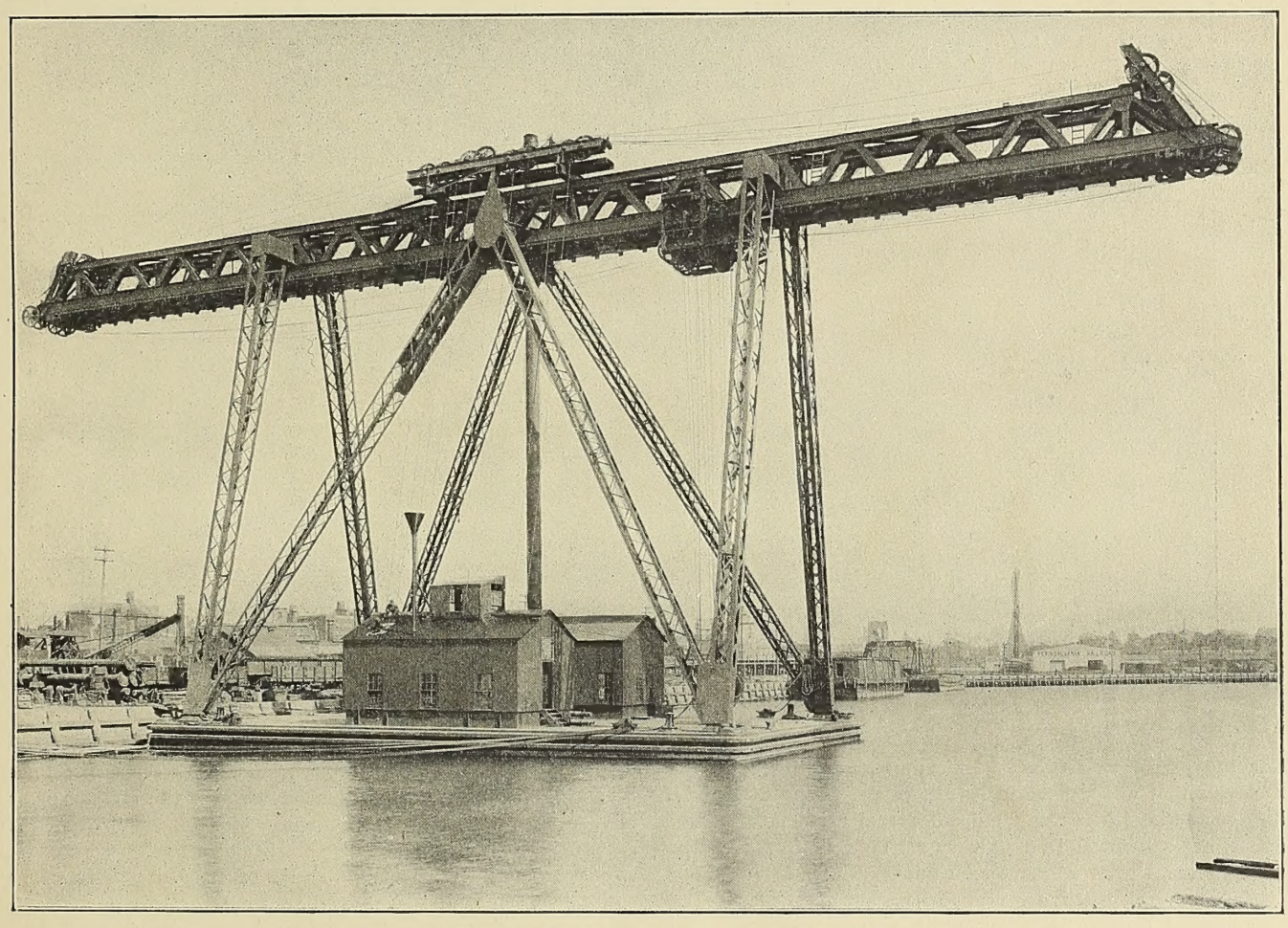
A NEW ONE-HUNDRED-TON FLOATING CRANE.

There is probably no piece of machinery so generally or so frequently used in a shipyard as a crane. From the time of the initial building of the ways and the laying of the keel to the swinging aboard of the last ton of coal for the trial trip, does this instrument of transportation play its important part in the construction of a ship. Its uses are varied and its forms are almost as diverse. A simple mast and jib on the deck of a barge or schooner unloads the material to cars on the wharf; a locomotive crane transfers it from these to the storage racks or piles; one of the traveling bridge type on elevated runways handles it in the work shop for shaping or assembling; and the elaborate cantilever structure mounted on tracks a hundred feet from the ground with booms extended fifty feet on either side, places the finished piece in its destined position on the vessel under construction. These, together with the stationary jib cranes and shear legs located on the wharves, are the kinds most commonly employed on land in shipyard work; but after launching there still remains a large portion of the vessel uncompleted, most of the material for which must be hoisted into place by means of cranes of the floating type.

The mast and jib is one of the earliest of engineering structures, but subsequent development was slow until the general introduction of iron and steel into all construction work, the increased size and weight of machinery, and the tremendous growth of bulky freight transportation, impelled a corresponding elaboration of means for rapid and economical handling. Thus resulted the complex and powerful locomotive and wharf cranes designed by the Germans, the labor-saving interior traveling bridge cranes of the American type, and the recent remarkable floating structures which are a necessity with the present method of finishing the major portion of the construction work on a vessel after it has been launched.



ONE-HUNDRED-TON FLOATING CRANE, FOR THE NEW YORK NAVY YARD, BROOKLYN, N. Y.

[Photograph by E. Muller.]

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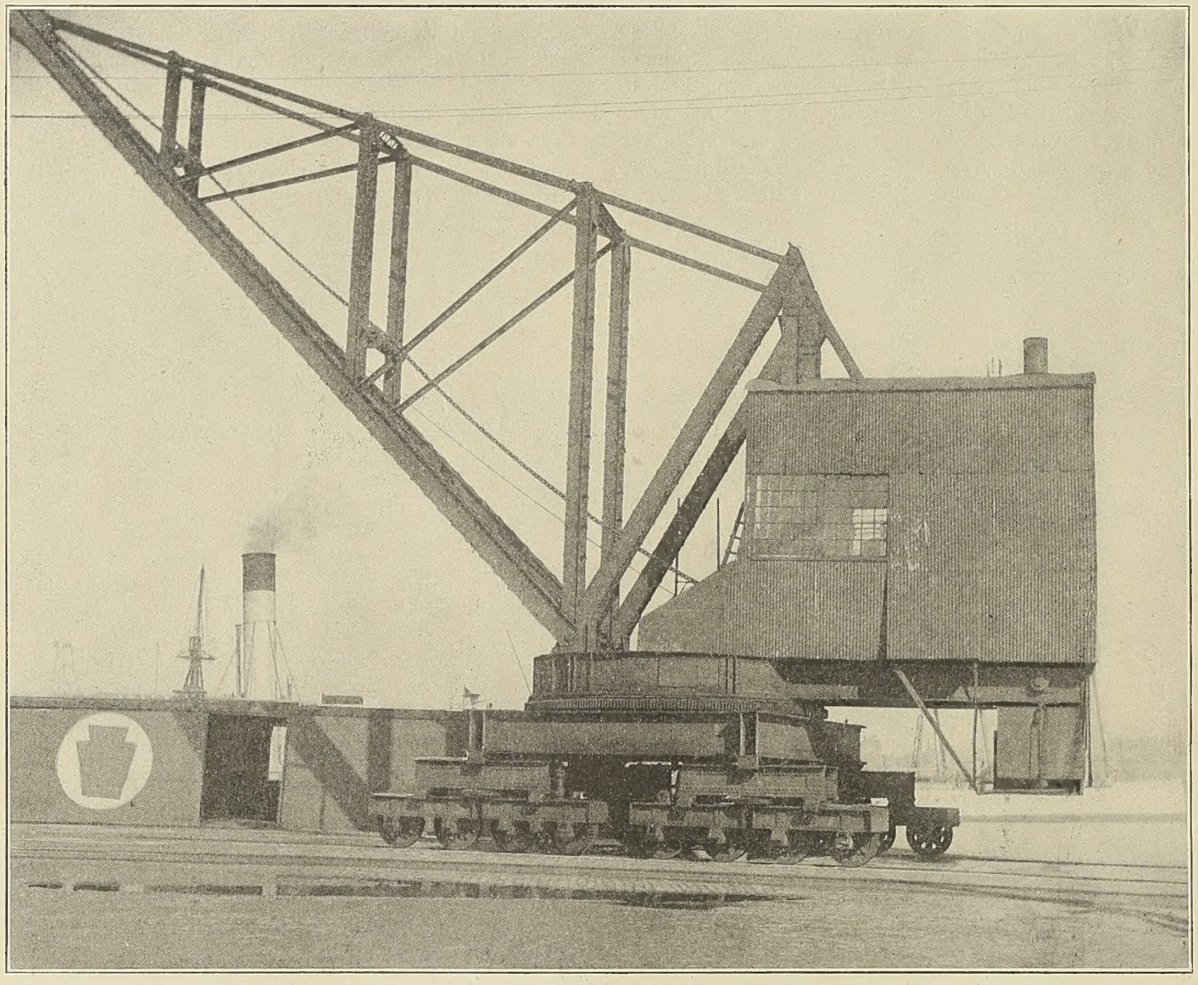
But the necessity for a crane of this kind is made absolute in a shipyard where heavy armor-clad war vessels are built and repaired, for only by such means can these operations be successfully carried on.

This latter fact was recognized by congress early in 1896, when the plans for the new navy began to assume tangible shape in the liberal appropriations for new vessels and navy yard improvements. It was not, however, until 1898 that a definite amount of money was set aside to be used for the construction of a steel floating crane of 100 tons capacity—the New York navy yard being designated to receive this machine, for then, as now, this yard did by far the largest amount of repair work and original building.



FORTY-TON LOCOMOTIVE CRANE, IN USE AT THE NAVY YARD, BROOKLYN

After the preparation of a specification giving minutely the requirements that would have to be met in the design of a crane of this type rather than the details of actual construction, proposals were asked for, and on January 23, 1899, the contract was awarded to the Brown Hoisting Machinery Company on their bid of $100,000. The delivery was to be made on February 12, 1900, but the problems arising as the design progressed were all new ones and were attacked with such thoroughness by the builders that it was not until July, 1903, that the immense structure was finally ready for test. On July 28, 1903, an exhaustive trial was made, under the supervision of Civil Engineer R. C. Hollyday, U. S. N., in charge of the engineering work at this yard, the machine being subjected to lifting, conveying, counterbalancing and manœuvring tests, the results of which were in entire conformance with the specification requirements.

Essentially this crane consists of a rectangular steel pontoon in which are installed the engines, boilers, and counterbalancing device, and the superstructure made up of the bridge for the trolley runway and the supporting struts.

The pontoon is 60 feet wide, 100 feet long, and 11 feet 3 inches deep, the whole being built of steel plates riveted to steel angles and channels properly braced, and it is divided by water-tight bulkheads into four main separate chambers, viz., two of them midway fore-and-aft, 18 feet long by 30 feet broad by 9 feet deep, for the engines and boilers; one 18 feet broad by 9 feet deep, in section, extending the whole length of the structure for the counter weight, and the remainder merely a water-tight space for buoyancy, which is further subdivided by six longitudinal and six transverse water-tight bulkheads to afford ample protection against sinking in case the hull be pierced through any accident.

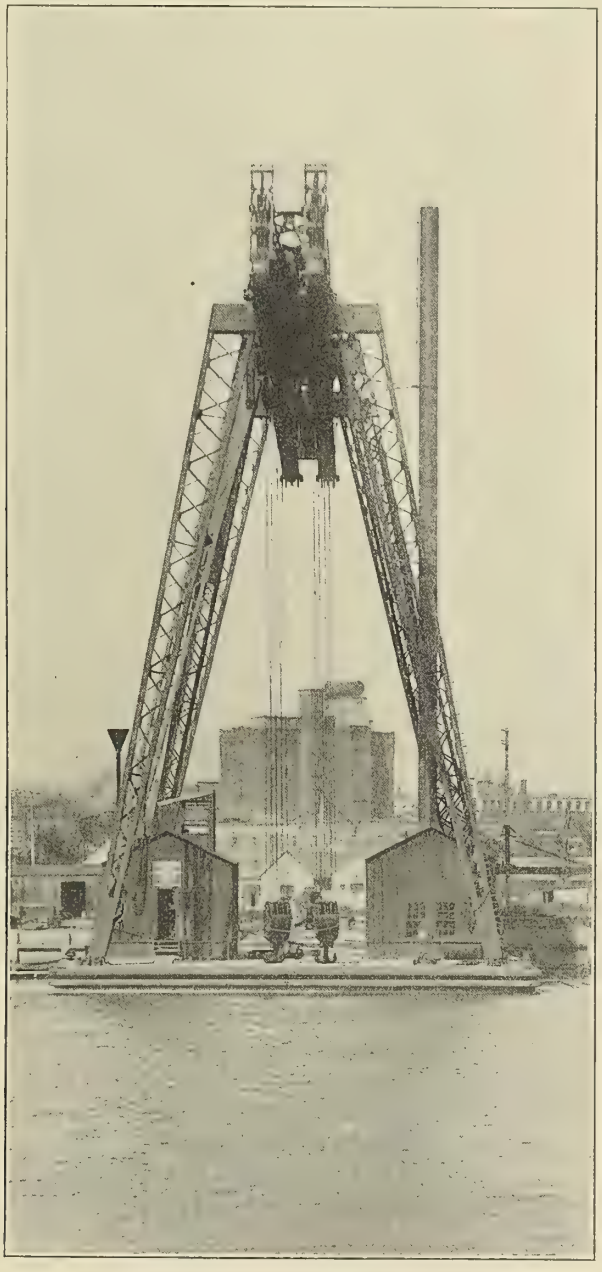
In the port side are located two hoisting engines of the horizontal type, with double cylinders 12 inches in diameter and 15-inch stroke, each operating through trains of three cast steel gears, four cable drums 5 feet in diameter, two for the racking or pulling ropes for moving the trolley and two for the hoisting ropes, each separately connected to a hoist block and capable of being operated independently. The drums are of such size as to take the entire length of rope in the grooves cut for this purpose without overwinding; they revolve on forged steel shafts, bronze bushed and provided with compression cup oiling devices; and they are controlled by a system of levers, located on the top of the engine room, by means of foot brakes and a friction clutch on each drum.

Here is also installed a vertical engine of the double-cylinder type, each cylinder connected through the crank shaft and a steel worm to a bronze worm gear actuating a shaft on which is mounted a cable drum, one for moving the counterweight aft and the other for moving it forward. This engine is controlled by an automatic balanced throttling mechanism operated by a device fastened to a float in a tank at each end of the pontoon, these tanks being filled with water and connected through their

bottoms by a pipe of a size sufficient to allow a perfect and rapid circulation from one tank to the other. Thus any deflection of the pontoon will cause a change of level of the water in the tanks, thus moving the floats, which in turn will admit steam to the engine and drive the drums in a direction proper to draw the counterweight in a position to balance the crane. This weight consists of a rectangular box heavy enough to counteract the maximum load lifted by the crane; it is mounted through springs on sixteen chilled faced charcoal iron wheels with double flanges and with axles turning in self-oiling journal boxes. These wheels rest on four lines of 60-pound rails, spaced 2 feet 6 inches on centers and extending the whole length of the pontoon. Independent means for controlling this counterbalancing device from the operating platform are also provided, and in case of any accident breaking either hoisting or trolley ropes, the weight is at once automatically locked in position by a separate control connected to the hoisting and pulling drums. That this counterbalance is effective was shown conclusively in the test referred to previously, when, due to newness, the lifting cables stretched, immediately locking the counterweight. At no time did the deck deflect more than 1 degree and 30 minutes from the horizontal, although 215,000 pounds were lifted and moved from one extreme end of the trolley bridge to the other.

In the starboard side of the pontoon are located two 100-horsepower boilers of the locomotive type, and the necessary boiler feed pumps, injectors, etc.

At each end of the pontoon a laced angle A-frame, pin-connected through a massive casting to the deck, rises vertically to afford support to the portion of the superstructure containing the trolley runway. Additional sustaining power and bracing against wind pressure are secured by means of two similar lateral A-frames riveted to the former at their bases and to the two open trusses of the trolley bridge immediately above the center of the deck. These parallel trusses are spaced 6 feet apart, center to center, are rigidly connected through their top and bottom chords by an elaborate system of bracing, and form on the faces of the two lower chords the requisite support for the lines of 60-pound rails extending the whole length of the structure, and upon which the trolley moves. These trusses are 218 feet 6 inches over all in length, the rails are 78 feet 4 inches above the deck, and the extreme travel of the center of the trolley is 45 feet beyond each end of the pontoon, a total of 190 feet.



END VIEW OF THE NEW ONE-HUNDRED-TON CRANE.

The trolley proper is made up of heavy plates and angles riveted together and hung through pin connections to the four trucks which are each mounted on four charcoal iron wheels, turning in self-aligning, bronze-bushed journal boxes. Within the frame work of this trolley are located twelve steel sheaves 4 feet in diameter, over which passes the hoisting rope sustaining the ponderous blocks for raising the load. Each of these blocks contains four sheaves 4 feet in diameter, so that eight strands of the hoisting rope, which is 1 1-8 inches in diameter, are used to carry the maximum safe weight of 134,400 pounds, for which each block is designed. The double terminal hook forming the attaching device on each block is of steel 9 square inches in section, and the weight hanging thereon is transferred to the block through specially hardened steel ball bearings having two circles of contact. The hoisting mechanism was purposely constructed in two separate parts, independently controlled, for thus can two connections be made to a large, heavy piece of material

REMOVING THE TURRET OF THE UNITED STATES MONITOR FLORIDA, AT THE NAVY YARD, BROOKLYN, N. Y.

such as a turret, a 13-inch gun, or an engine, and the level governed with absolute accuracy when lowering into position. This feature is of great importance, for in this manner entire contact is made simultaneously in cases where a line or a point contact would ruin the piece being handled.

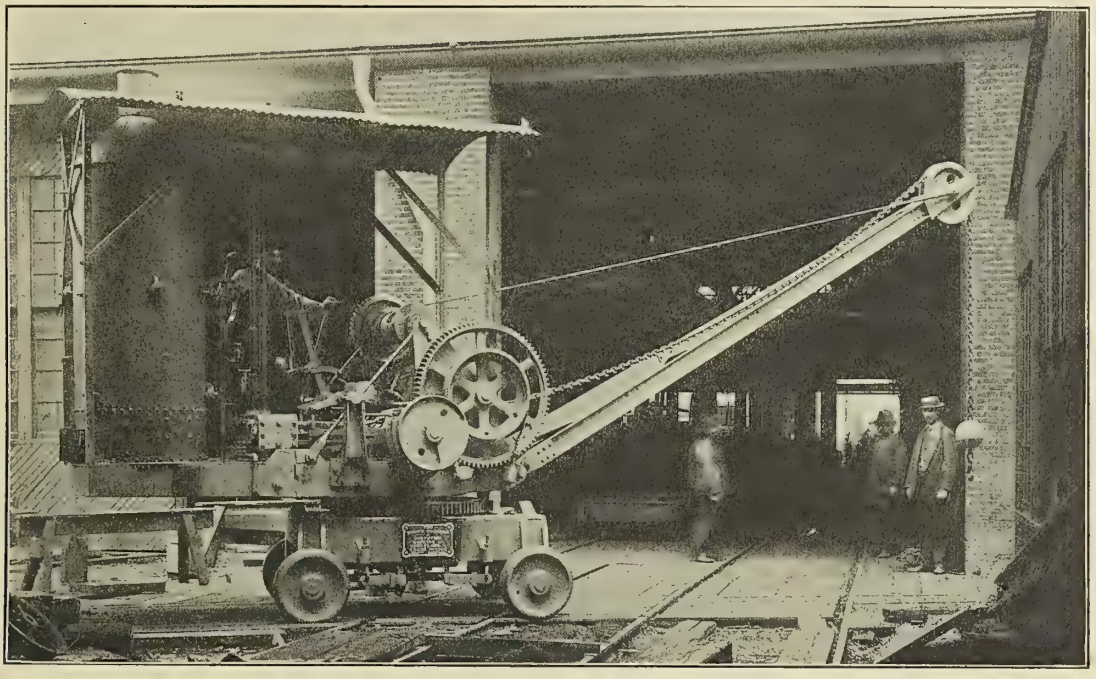
As above stated, the limit of horizontal travel of these blocks is 190 feet, and they are capable of being raised 64 feet above the deck and 22 feet below it. The speed is 54 feet per minute for vertical hoist and 106 feet for horizontal travel. The racking and lifting ropes are of the best plough steel wire, 1 1-8 inches in diameter, the ends of the latter being secured at either extremity of the trolley bridge to an equalizing device. There are in all 5,565 feet of this rope, guided on the crane by 48 steel sheaves 4 feet in diameter, practically all of which is in use when the blocks are in their lowest possible position with the trolley at the end of the bridge.

In addition to the maximum load on the hoisting mechanism, there can be simultaneously borne on the deck a weight of 300 gross tons with 2 feet freeboard, which thus allows a large amount of material to be removed from a vessel and stored on the deck prior to final transportation to the shops and storehouse. The total weight of the crane itself, which is of open hearth steel throughout except doors, windows, and the fender strips, is 2,656,000 pounds.

It was originally contemplated to equip this machine with propellers so as to make it an absolutely self-contained structure, but this was found to be wholly impracticable owing to the immense size; and removal from place to place is now done by tugs and the final warping into proper position by four winches, one at each corner, driven by a 5-inch by 8-inch double cylinder steam engine secured to the under side of the deck.

Even the projectors of this crane could not foresee the great usefulness of a machine of this character; and it has been found perfectly adapted to the diverse kinds of work on which it has been used during the fifteen months. Originally intended for handling heavy and bulky vessels' equipment, the requirements in this regard have been more than met; for with it immediately upon acceptance, the great armored turret of the U. S. monitor *Florida*, which weighed 120 gross tons, was successfully removed and replaced, and since then the turrets and heavy guns of the battleships *Iowa* and *Indiana* have been handled equally as well. The most interesting confirmation, however, of its ready and economical adaptability to this class of work was seen recently when it was taken to Jersey City and used to unload a new 13-inch gun from the specially designed railroad car to its deck, where it was conveyed back to the navy yard. Here, after removing a defective gun, the crane placed the new one in the turret of Admiral Evans' former flagship, the *Kentucky*, with absolute precision and without the slightest accident. To do this work a private concern asked a compensation of $4,000; the saving is astonishing, for the amount actually expended was less than $200. The economy resulting from its use is not alone confined to operations such as these, for in the building of a new

pier at the New York yard from a large quantity of immense granite blocks formerly used for quay-wall construction and which had to be resurfaced, this crane was utilized. An air compressor was installed next to the engine room and a large pneumatic surfacing machine was placed on rails on the deck of the pontoon. The hoists furnished ready means for the required handling of the granite and the dressing was carried on simultaneously with the other work in which this crane was engaged. Another very important case in which this giant machine proved to be extremely useful was in the removal of an old granite and concrete quay wall on pile foundations which extended across the ways on which the U. S. S. *Connecticut* was building. By this means the masonry portion was taken out in large pieces and the piles were then removed whole without injury. At this time an incidental demonstration was made by displacement observations that the resistance to upward pulling in piles varies from 50 to 75 tons per pile in ordinary soil, which data are very valuable in connection with the design of dry docks where there is an immense upward pressure due to buoyancy to be overcome when the dock is empty.



TEN-TON LOCOMOTIVE CRANE AT THE BOSTON NAVY YARD.

An additional fact regarding this crane is appropos here and will probably further confirm the admirable foresight of the naval engineers who urged its construction. Early in 1903, when war clouds were descending rather thickly in Asia, and Europe as well, and when this crane was about ready to be turned over to the government, a representative of a foreign state, duly accredited for the purpose, offered $250,000 for this machine, which tender was, of course, refused. The wide range of usefulness of this absolutely unique structure, in point of both size and design, was thus reserved for the navy of the United States, where its importance will grow as the plans for our new navy mature, and where, if war occurs, it will prove one of the most valuable adjuncts in the repair and equipment of our naval vessels.

Monthly Shipbuilding Returns.—The Bureau of Navigation reports 83 sail and steam vessels, of 20,282 gross tons, built in the United States and officially numbered during the month of November, 1904, as follows: 37 sailing wooden vessels, of 16,859 tons; 40 wooden steam vessels, of 1,680 tons; one steel sail vessel, of 331 tons; and five steel steam vessels, of 1,412 tons.

IN THE ENGINE ROOM OF THE STEAMER TURBINIA.

BY MORRIS M. WHITAKER.

The writer had the opportunity some few weeks ago of inspecting the first turbine-driven vessel in commercial use in this hemisphere, on her daily run between Toronto and Hamilton, Canada, and was able, through the courtesy of the owners, to obtain particulars which may be of interest to the readers of MARINE ENGINEERING. The *Turbinia,* built by Hawthorn, Leslie & Company, and engined by the Parsons Marine Steam Turbine Company, has fulfilled her owners' expectations by a very successful season. The passenger and freight traffic between Hamilton and Toronto during the summer season is very large, and up to the advent of *Turbinia* had been divided between the railways and a line operating two steamers of about 14 miles speed, with the latter carrying the bulk of the traffic.

Some enterprising gentlemen of Hamilton determined to enter this field, and with characteristic energy and forethought decided to take advantage of modern progress in marine engineering and use the turbine principle. They accordingly placed an order with Parsons about the last of December, 1903, and with the opening of the season of 1904, the *Turbinia* was making her bid for popular favor. Her speed, steadiness, absolute regularity of service, and, possibly, novelty, have made her a favorite with the public; while her economy in running expenses, such as attendance, coal, oil, etc., have proved her a success from the owners' standpoint. As she is fully equipped for use in salt water, it is the intention of the owners to fill in the closed season on Lake Ontario by running her between Nassau and some Florida port, thus having practically no break in her earning capacity.

The theory of the turbine has become fairly familiar nowadays, but very little has been published giving information from those who have had close practical experience with the new system of using steam, and accordingly I will try to give some account of it as gathered from the engineers in charge of *Turbinia's* machinery, and personal impressions from watching it in operation.

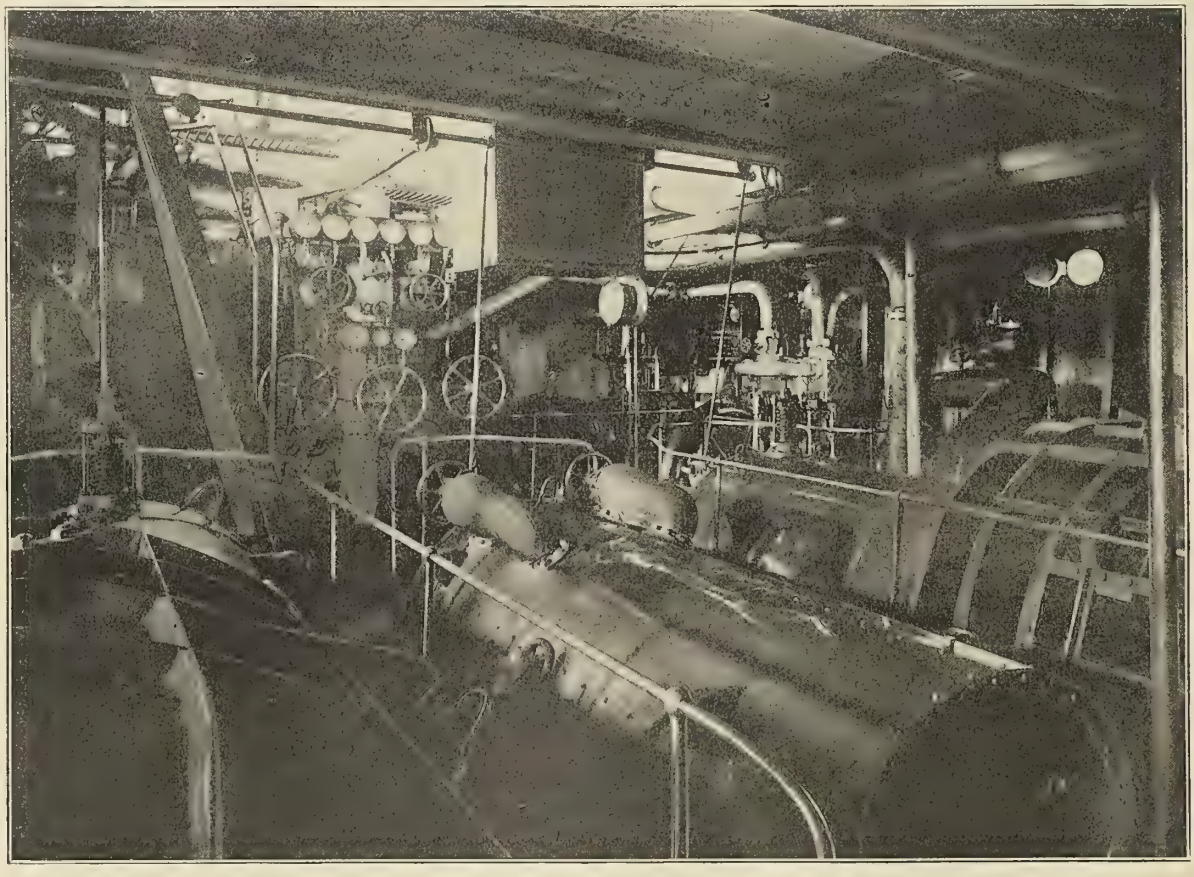
After her trials on the Tyne, the vessel started across the ocean, and, although she had a very rough passage, the run from Stornaway to Sydney, Cape Breton, was made in six days, at the rate of 17 1-2 knots, head seas being the rule rather than the exception. Although *Turbinia* has very fine ends and pitched a good deal, the governors never came into action, even though they were set at only 40 revolutions above the normal. Her coal

capacity in bunkers is 110 tons, and for the trip 500 tons additional were taken aboard, and yet, on her arrival at Sydney, she had over a day's supply still on hand, which is very good considering her rough trip and that she was not eased off on account of the weather. I believe she is the first turbine-propelled commercial vessel to cross the ocean.

As is usual, the propelling machinery consists of a high-pressure turbine placed on the center shaft and two low-pressure turbines placed in the wings, each exhausting into its own condenser. The circulating and air pumps are in duplicate, as are the feed pumps and the oil circulating pumps for the bearings on the turbines. The arrangement of the machinery is shown below, and occupies a fore-and-aft space of about 30 feet.

The attention required by the engine-room force after once starting is very small, and most of this is devoted to the auxiliary machinery. The watch consists of one engineer, one oiler and water tender, three firemen, and one coal passer, to take care of machinery of 3,400 estimated horse power and all the auxiliaries. The oil placed in the oiling system when she was tried on the Tyne is still being used, and the main engines have never been even opened out for examination, nor has any adjustment been made during the season. It is a pleasure to watch the manœuvring possible with this type of machinery, and how beautifully it is made to respond to orders from the deck. In starting and coming into the dock two of the engine-room force are stationed at the three operating wheels shown in the cut of the engine room. The center wheel controls the admission of steam to the high-pressure turbine (which is used only when going ahead and not in manœuvring) and those to starboard and port the admission to the wing turbines. By turning these wheels to right or left, steam is admitted directly to either the go-ahead or backing ends. The control is practically instantaneous, and so smooth and vibrationless that if one did not watch the governor balls on the forward ends of the turbines and hear the rush of the steam in the main steam pipes, he would not be aware of any movement. The slow, half, or full-speed, ahead or astern, are given by the amount of steam admitted, and can be regulated by the steam gauges directly in front of the operator.

When the boat is once clear of the dock and on her course, the valve to the high-pressure is opened and those controlling the admission into the wing turbines are closed, and the steam takes its course through the high-pressure center turbine to the low-pressure wing turbines. When running throttled down there is considerable noise due to the rush of steam through the throttle, but when the valves are full open this is not noticed and there remains only the roar of the auxiliary machinery. Amidships no vibration is perceptible; but at bow and stern, and especially the latter, considerable vibration is felt, which is doubtless due to the propellers. If the auxiliaries were run on the turbine principle, a person standing amidships would hardly know any machinery was in motion. When running at full speed *Turbinia,* due to her beautiful model, makes no more fuss than many boats of similar size at half the speed.



ENGINE ROOM OF THE CANADIAN TURBINE STEAMER TURBINIA, LOOKING FORWARD.

The following are particulars of trial on Lake Ontario shortly after arrival:

Boiler pressure	160 lbs.
High-pressure turbine	122 lbs.
Low-pressure turbine	45 lbs.
Vacuum	27 1-2 ins.
Revolutions center turbine	650
Revolutions wing turbines	717

Speed, 31 1-2 miles (5,280 ft.) in 1 hr. 20 minutes.
Coal per hour 2 2-10 tons or 1.46 lbs. per I. H. P. per hour.
Feed temperature ..... 178 degrees
Evaporation, 8 lbs. water per lb. coal.

On regular runs boiler pressure is maintained at about 135 pounds, steam at high pressure 90 pounds, low pressure 32 pounds, vacuum 27 inches, revolutions center turbine 550 to 575, wings, 600, speed 31 1-2 miles 1 hour 25 minutes to 1 1-2 hour.

The high-pressure turbine on the center shaft turns right-handed, and is fitted with seven steps of blades with from 5 to 7 rows each. The diameter of the turbine is 40 inches, and that of

the outside casing 48 inches; the length is 8 feet 6 inches. The two low-pressure turbines, one on each wing shaft, have five steps of blades and are 48 inches diameter, outside of casing 56 inches. The length of the turbine is 11 feet, which is made up of 5 feet 6 inches for the go-ahead machine and 5 feet 6 inches for the backing turbine; thus there is a backing turbine in the casing of each low-pressure turbine. The blades vary in length from 1 1-4 to 6 inches and have a clearance of 1-32 inch. The governors and collar thrusts are placed at the forward ends of shafts. All shafting is 5 1-2 inches diameter. The propellers are solid bronze, elliptical shape, true screw blades, 49 inches diameter and 44 inches pitch.

The weights of machinery are as follows: Each of the two boilers, with water, 70 tons; the three turbines and shafting 58 tons; the auxiliary machinery 5 tons. Each low-pressure turbine exhausts to a condenser next to the skin of the ship and each condenser has its set of air and circulating pumps, which are driven by a pair of reciprocating engines 9 by 7 inches, running at a speed of 200 revolutions per minute. The circulating pump is placed at the forward end and the air at the after end. The feed pumps are in duplicate and of the long stroke, slow-moving, Woodson type, 28 strokes per minute. A pressure of 5 pounds is maintained in the oiling system by two Weir pumps.

Air pressure on the closed stoke-hold system is maintained in the fire room by a double-acting, high-speed blower engine, direct connected to fan. The ship is lighted throughout by electricity by means of a direct-connected generating set.

MARINE ENGINEERING DATA CARD. JANUARY, 1905.

NAME—Turbinia. NATIONALITY—Canadian. TYPE—Lake passenger steamer.

LAUNCHED—1904. BUILDERS { Hull and Boilers—Hawthorn, Leslie and Company. Turbines—Parsons Marine Steam Turbine Company.

Length over all	260 ft. 0 ins.
" between perps	250 " 0 "
Breadth, molded	32 " 6 "
Load draft, mean	9 " 0 "
Load, displacement	1100 tons
Block coef.	.53
Engines:	Parsons turbines
One high press., on center shaft turn right hand.	
Two low press., on 2 wing shafts turn right and left.	
Expansions, total number	127
Horse power, estimated	3,400
Shafting, main and tail, 5 1-2 ins. diam	
Propellers:	
Solid bronze casting.	
True screw.	
Diameter	49 ins.
Pitch	44 "
Boilers:	
Two single end Scotch.	
Length	10 ft. 6 ins.
Diameter	17 " 6 "
Pressure, gauge	160 lbs.
Furnaces: each	4 Morison
Diameter	42 ins.
H. S.	6,688 sq. ft.
G. S.	182 " "
Draft; forced, closed stoke hold.	
H. S. ÷ I. H. P. (estimated)	1.97
H. S. ÷ G. S.	36.7
I. H. P. (estimated) ÷ G. S.	18.7

Florida Motor Boat Carnival and Races.—The first annual motor boat and water carnival at Palm Beach, Fla., will take place February 1 to 3 inclusive. It is proposed to give four races each day. The races will take place on Lake Worht, a fine sheet of water, and it is hoped that many of the representative builders and owners will enter boats for this event, which is under the Palm Beach Power Boat Association.

PRODUCER GAS ENGINES ON SHIPBOARD.

BY BERNHARD A. SINN.

The engine room equipment of a vessel propelled by gas is much the same as that of steamships. The machinery of steam vessels may be divided into two general subdivisions: the driving engines and the auxiliaries. The latter may be subdivided into four classes: those auxiliary engines necessary for the proper operation of the main engines, those required for the boiler room equipment, such machinery as is needed to keep the vessel itself in good running order and to assist the loading and discharging of the same, and, lastly, those required for the convenience of the passengers and crew.

In the operation of the main engine's condensers, air, and circulating pumps, water and oil circulating pumps are necessary.

For the proper operation of the boilers, feed pumps, distillers, evaporators, blowers and fans, ash ejectors, superheaters, feed heaters, injectors, etc., are needed.

For the proper operation of the vessel itself we must have windlass, steering gear, bilge, ballast and fire pumps, and where cargo is carried, winches and other deck machinery. Those carrying dressed beef, fruits, etc., must have a large refrigerating plant.

For the convenience of the passengers and crew electric light machinery, heating and ventilating systems, sanitary and fresh water pumps, distilling apparatus and hot water supply are required.

In a vessel propelled by gas the latter two subdivisions are necessary in all their details, and in place of the first and second, other machinery has to be installed to accomplish such purposes a the requirements may call for.

A gas propelled vessel requires in addition to the main driving engines exhausters for the gas producers; an air compressing plant to reverse the main engines, operate other engine gear, and to blow the whistle; circulating pumps for circulating water through the main engines and to give water to the steamer of the producers; and other auxiliaries not directly used for the propelling engines, such as re-heaters for heating and sanitary purposes. In other words, the amount of auxiliary machinery necessary in a gas propelled vessel is no greater than that in steam vessels and much less complicated.

The manufacturers of pumps are now supplying electrically driven pumps which have a high efficiency as far as power is concerned, and are very much more economical than direct driven steam pumps, and as it is a well known fact that the modern steam pump is a most uneconomical piece of mechanism, it is possible to generate auxiliary power cheaper with producer gas and gas engines and supply electric power from this source than would be the cost of running auxiliaries by steam direct.

Where a gas producer plant is already installed on board ship, a direct driven electric plant of sufficient size to furnish power for the auxiliary machines and to do other work should be installed. The compressed air is furnished by a small electrically driven air compressor for the start, and, if desired, from the main engines when running. These machines are now built so that the power is regulated by the pressure and practically a constant air supply is furnished.

It is unfortunate that the high price of liquid fuel has prevented the adoption of gas engines for large powers on board ship. Although in recent years some vessels of considerable tonnage have been equipped with oil engines and proven very successful, the main objection to oil engines heretofore has been, firstly, the higher insurance rate demanded, owing to the popular belief that oil fuel is dangerous to handle, and, secondly, the non-reversible properties of the engine. In order to drive the vessel astern it is customary to use a combination of heavy gearing which will give opposite motion to the shaft and propeller. This is not objectionable for small powers, but it is manifestly impracticable to design a reverse gear to transmit large powers. If it were possible to design a gear that would reverse an engine of 5,000 I. H. P. having an 18-inch crank shaft, it would require gears of 12 feet diameter and 4 feet face. There is in addition always the danger of a

breakdown in the reverse gears through defective teeth and castings and slipping of the clutches and dogs.

Manufacturers of marine gas engines in the last two years are, however, building engines that may be reversed by compressed air. This is accomplished by cutting off the gas, turning the cam shaft 90 degrees, and sending compressed air into the cylinder when the change in the time of admission and exhaust is sufficient to catch the engine in the reverse motion in one of the three cylinders and reverse the same. As soon as the engine is started in the opposite direction gas is admitted through the proper valves, the time of ignition arranged to suit, and the engine operated in the reverse direction in the same manner as previously in the go-ahead motion. All this is accomplished with one lever and valve, the lever operating the cam and ignition shafts, and both of the compressed air and gas valves.

Although no engines of considerable size have been installed aboard ship owing to the high price fuel, as before mentioned, the introduction of the gas producer will soon compel progressive vessel owners to give the matter deep thought. It is therefore necessary to turn to land practice for such information regarding gas engines in order to speak intelligently on the subject. Both in this country and abroad vertical gas engines of 3,000 to 5,000 horse power have been successfully installed for the production of electric and other power. These engines have all the requisites of a marine engine excepting reversibility, and this point can be readily taken care of as before mentioned. This is a good starting point for the designer of marine gas engines, as there is a definite piece of mechanism from which to draw. Manufacturers of large gas engines on land are building double-acting engines of both the four and two-cycle types.

For small powers a two-cycle engine is preferable in a great many cases, owing to its light weight and the fact that it can be reversed by hand, whereas a four-cycle engine cannot; but in large powers, particularly on shipboard, where space is a great consideration, the lack of economy of the two-cycle engine and its very much increased height, would be more than sufficient to offset the increased weight of the four-cycle type. Taking these things into consideration, it is certain that the four-cycle engine, as it is being built to-day in large powers, is the more desirable type and can be safely recommended for marine purposes. A three-cylinder, double-acting, four-cycle engine gives six explosions in each two revolutions, which is the same as a two-cycle, single-acting engine.

In the best gas engine practice the following proportions are generally adopted:

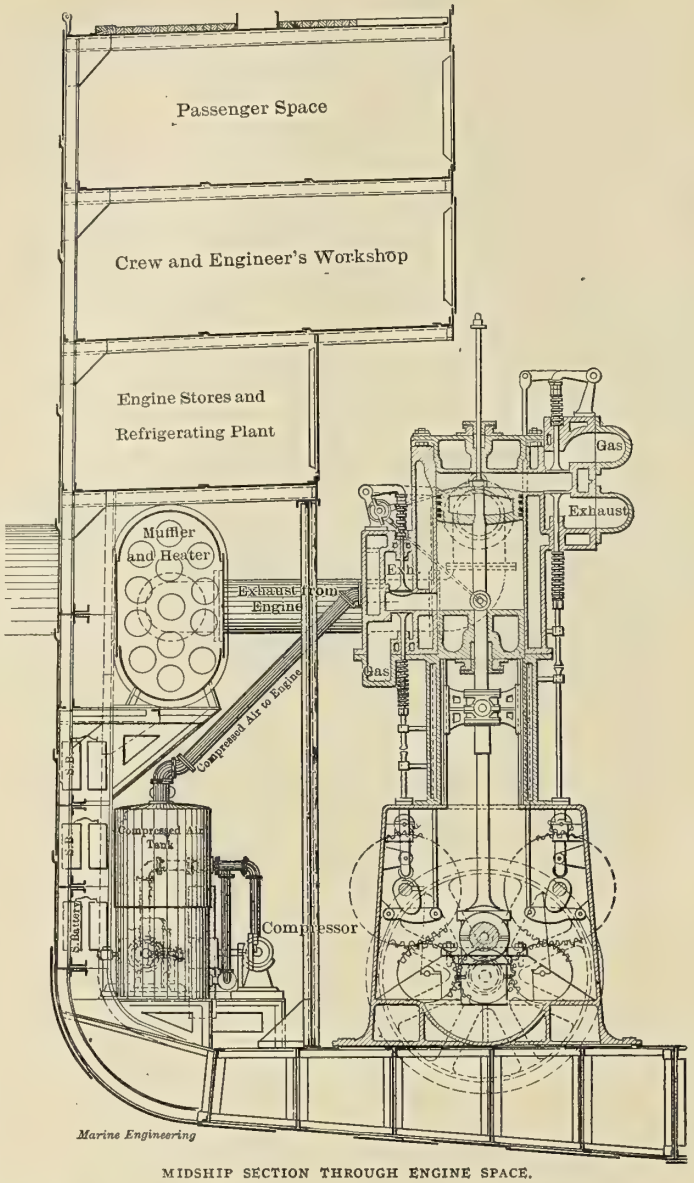
With producer gas and a well-designed engine operating on a high compression, 85 cubic feet of gas will prove sufficient for one indicated horse power. A piston speed of about 800 feet per minute is as high as has yet been attempted, and a mean effective pressure per explosion of 70 pounds is generally obtained in good practice, although as high as 80 pounds per square inch has been shown. These considerations determine the size of the engine cylinders.

The speed of 100 feet per second for inlet and 85 feet per second for the exhaust gases are generally accepted figures. It is necessary therefore to design the valves and ports of such areas as will admit and exhaust the gases at these speeds. The valves should have an area of 15 to 20 per cent. of the cylinder area, and should give a quick opening and extremely rapid close. With producer gas a mixture of about equal quantities of air and gas gives the total volume of mixture necessary, and with these proportions the valves may be readily designed of proper size.

The compression space for a compression of 150 pounds per square inch should be 1-4 the volume of the piston displacement, and the height of the compression space 5-8 the diameter of the inlet valve.

With a double-acting engine, piston and connecting rods are necessary; in single-acting engines the piston rods may be dispensed with.

The cam shafts and ignition shafts are directly operated by



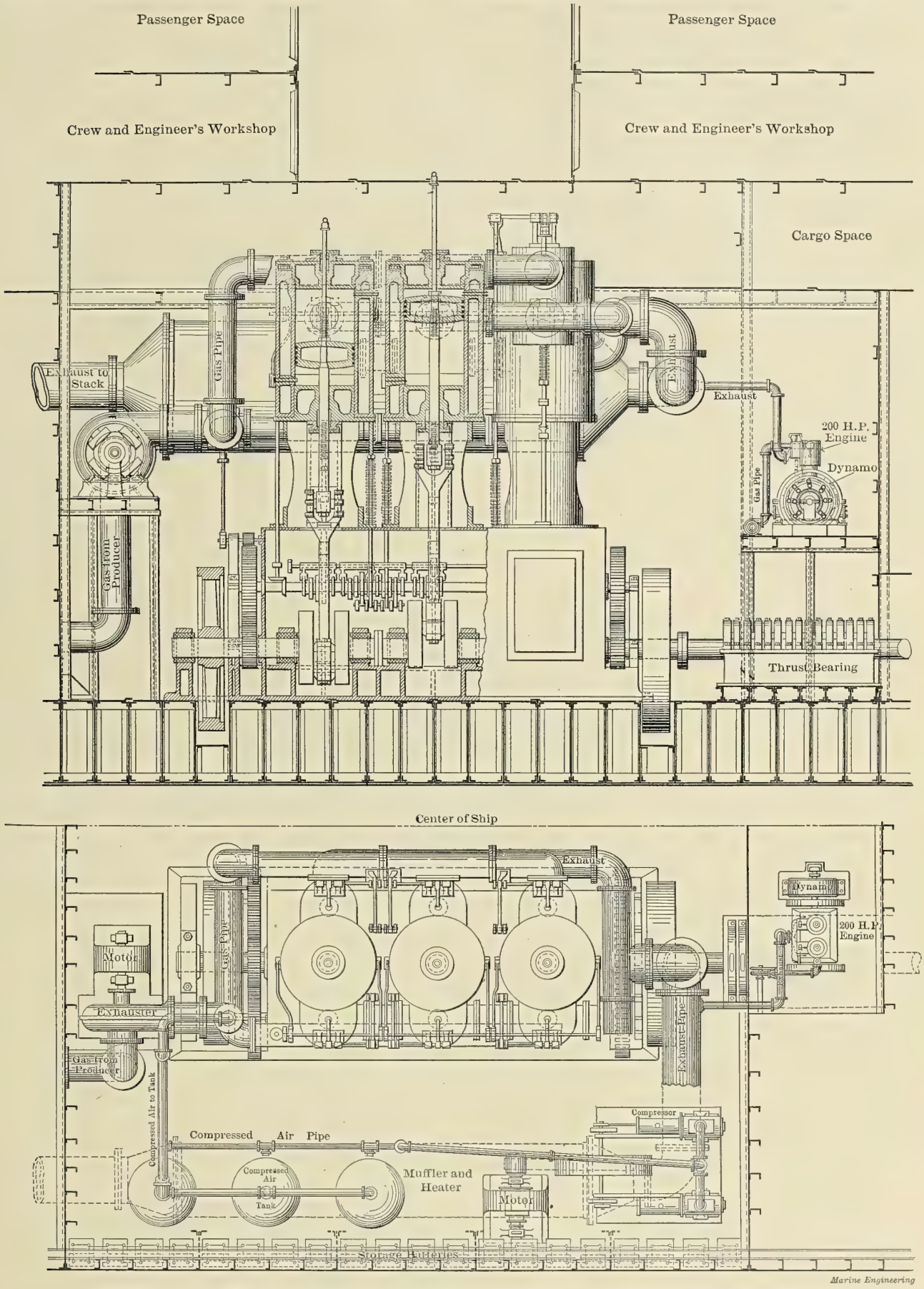
MIDSHIP SECTION THROUGH ENGINE SPACE.

gearing from the main shaft and the cams are designed to give the proper lift of the valve at the proper time. For each valve there will be three cams—one for operating the valve when the engine is going ahead, one for operating the valve when the engine is being reversed, and the third for operating the valve while the engine is going astern. This is accomplished by making the cams movable on the cam shaft and by means of forked levers pushing them forward and backward on the shaft. A small air cylinder and rack will do this, and a locking device is added for permanence in either the ahead or astern motion.

The ignition is timed by means of a counter shaft, with the usual make-and-break hammer spark arrangement; this counter shaft is operated by gearing from the cam shaft. When the cams move horizontally while the engine is being reversed, the time of ignition is changed at the same time by twisting the shaft through the proper arc and thus changing the time of spark.

The bearings, connecting rod, crank shaft, piston rods, slippers, guides, cross heads, etc., are all of the same type as in a steam engine of equivalent size. The piston is somewhat different, generally convex on top and bottom and hollow, and is ground into the cylinder. The piston and tail rod stuffing boxes are exceptionally deep and have very heavy packings, to prevent them blowing out from the high compression and explosive pressure of the engine, and to prevent leaking in any way whatsoever.

The cylinders, ports, valve chests, exhaust valves, as well as the



ELEVATION AND HALF-PLAN OF ENGINE ROOM, WITH TWO 5,000-HORSE-POWER GAS TURBINES FOR USING PRODUCER GAS.

pistons, tail rods, and connecting rods, are water cooled, the water being supplied by means of an independent circulating pump or a direct driven circulating pump from the engine itself.

The materials used in the construction of an engine of this type must be of the very strongest and best; cast steel is preferable in all castings excepting the cylinders, which should be of very fine, close grain iron; the valves should be of steel ground into place, the valve stems, tail rods, and piston and crank shaft

of the very finest quality of forged steel hollowed where possible to lighten weight. The valves should be all operated by direct springs, which must be kept constantly oiled and should be incased to prevent dirt and dust getting into them. Mufflers should be placed around all openings from the cylinder that would be likely to transmit sound, and cases should also be placed over all ignition plugs and heads. These cases should be made in tin lined with asbestos or other non-conducting material.

The gases after being burned in the cylinder of the engines are led to a muffler where the sound of explosion is reduced to a minimum and the heat in the exhaust gases utilized for raising the temperature of water for hot water and heating purposes; in this way such heat as is required for the vessel is procured from the exhaust at practically no cost, the gases being led to the stack after their passage through the muffler. When the engines are not running gas may be directly burned under the heater or muffler and the water heated by this means. An independent pump supplies such water as is necessary, and still further economy is obtained by using the jacket water of the engines—which is also used in the gas generators for steam purposes.

The compressed air plant is necessary for reversing the engine and blowing the whistle. It consists of two independent units storing air in tanks at a high pressure and then piped to the reversing valve and whistle. This air compressor is operated by a motor and can be run at a low or high speed, as desired, and may be so arranged that the motor is started or stopped electrically from the pressure gauge.

The most important auxiliary plant in a gas propelled vessel is the electric plant, which must be capable of doing a large variety of things. As soon as the starting producer is supplying gas the electric plant is ready for operation. It must store up energy in the accumulators; be ready to operate such pumps as require to be used; drive the blowers for starting the main generators, and also the exhausters for supplying gas to the main engines; it must run the various small pumps throughout the vessel, the deck machinery, steering gear and windlass, and supply light as needed, and supply the ignition spark for the engines. For this purpose in the illustration there are shown two plants each of 150 Kw., capable of being overloaded 50 per cent., and this 300 Kw. in addition to such excess as may be derived from the storage batteries, will be sufficient to operate the full auxiliaries of the vessel. The dynamos are connected to the gas engines and the piping so arranged that the gas engines may take gas either from the starting machines or from the main producers. As the storage batteries also are connected to the main switchboard the motors may be operated from generators or from storage batteries.

The main and other engines should be supplied with oil from tanks with automatic filling devices, and means for re-using the oil in the crank pit, etc., should not be neglected.

The two main driving engines with their thrust bearings, as is customary in twin-screw vessels, occupy the center of the engine room space. On each side on the floor are air compressors and tanks and such pumps as are necessary. On platforms built at the forward end of the engine room are the exhausters which suck the gas from the gas generators and deliver the same to the engine. On a platform at the atfer end of the engine room are the dynamos. Above the hold stringer on each side are the mufflers and hot water heaters, and in the lower 'tween decks the refrigerating plant, machine shop and other small auxiliaries. The storage batteries are placed well out at the side between the frames and on the stringers. The blowers for starting the generators are also situated in the engine room.

The piping is so arranged as to be easy of access, and one noticeable feature is the absence of all the small steam piping that is so difficult to place in steam vessels; in place of this there are only the electric wires and the main gas, air, and exhaust pipes, the auxiliary gas pipe to the electric generating sets, and the piping from the air compressor plants.

This description will in general describe the requirements of a gas propelled vessel; it is difficult at the present moment to say what means will be employed for accomplishing these different results, but no doubt very little radical departure from present methods will be utilized for some time to come.

In the comparison of steam and gas the three things which must be taken into consideration are, first, cost of operation; in this may be included first cost in the form of interest, depreciation, and insurance; second, weight of machinery and accessories; and, last, the space occupied. The value of a vessel is its earning capacity, and this must not be sacrificed in any form unless the cost of operation and management would far outweigh these considerations.

If the machinery of a steam vessel of size such as has been employed for comparison in this article is divided into groups, the first group containing boiler room equipment, the second the engine room equipment, the third the auxiliary machinery, and the fourth fuel and water, and a comparison made with the gas propelled vessel, it will be found that there is very considerable difference in weight, group for group. The boiler room equipment of a steamer weighs more than the producer room apparatus on a gas propelled vessel; the gas engines, however, are somewhat heavier than the steam engines would be, but if the necessary bunker coal and water in both vessels be taken into this comparison it is apparent that the economy in coal consumption more than balances any consideration of weight in the machinery itself. As the thwartship bunkers may be dispensed with, the total length of the engine and boiler space is eight frame spaces less in the gas propelled vessel than in the steamer, and this twenty feet thrown into cargo space gives just sufficient cubic capacity to stow the additional cargo that may be carried, owing to the decreased weight in machinery and bunker coal.

In the small tabulated statement a comparison of costs and weights for a 10,000 I. H. P. plant is made, and a decided saving in first cost, in operating expenses, in space, in weight, and an increase in carrying capacity and a large increase in earning capacity are shown, so that the future of gas propelled vessels is assured as soon as engine builders shall supply proper machines.

WEIGHTS FOR 10,000 I. H. P. MARINE ENGINES.

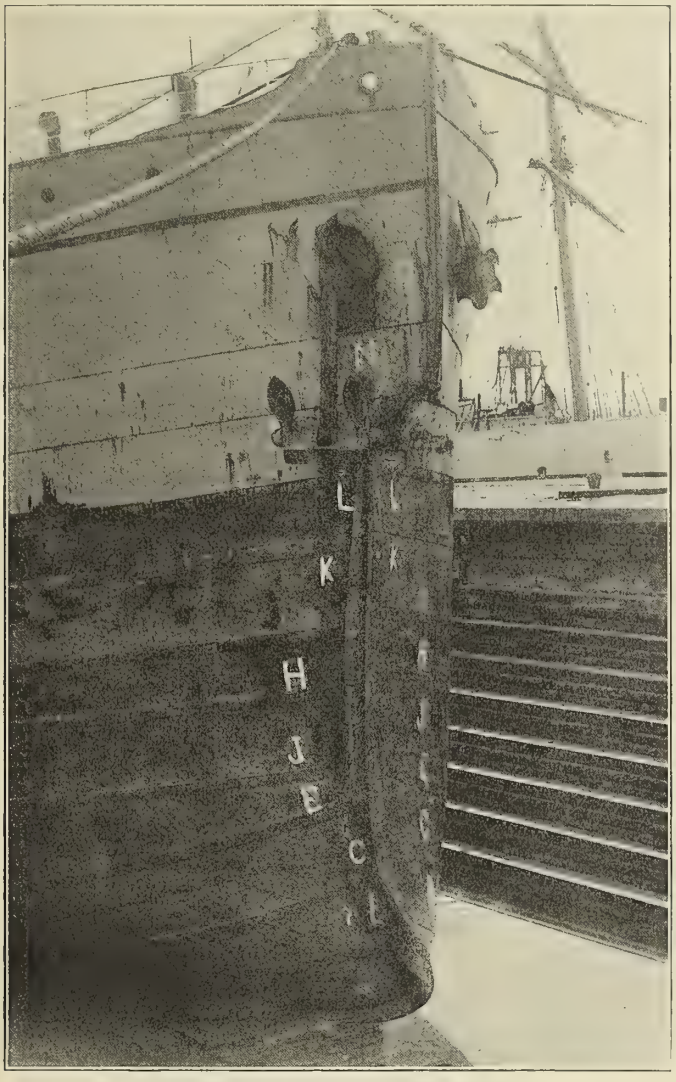
Steam Power Vessels.	Tons.	*Gas Power Vessels.*	Tons.
Main engines	385	Main engines	465
Condensers	56	Heaters and mufflers	32
Boilers	315	Generators	165
Uptakes, stacks, etc.	86	Relief and exhaust pipes	38
Fans, feed heaters, etc.	41	Exhausters and blowers	36
Pumps and auxiliaries	19	Compressor air plant, pumps, etc.	66
Electric light plant	20	Electric plant	40
Spares and sundries	15	Spares and sundries	18
Donkey boiler, complete	32	Starting machines	22
Water in boilers	180	Water in generators, etc.	45
Total	1,149	Total	927
Coal	1,950	Coal	1,000
Grand total	3,095	Grand total	1,927
First cost	$295,300.00	First cost	$246,200.00
Cost per I. H. P.	29.53	Cost per I. H. P.	24.62

	Steam power vessel.	Gas power vessel.
Interest, depreciation, and insurance	$27,620	$21,480
Coal, annually (150 days' steaming)	90,000	48,500
Service (engine and fire room crews)	30,000	12,000
Oils and stores	3,650	5,600
Repairs and sundries	15,000	12,000
	$166,270	$99,580
Annual saving	$66,690	
Added cargo capacity 1,150 tons. Value of added cargo carried	46,000	
Total saving	$112,690	

being about 10 per cent. on the whole cost of the vessel.

A great many gas engineers of prominence are to-day working on the gas turbine, and while it is yet too early to speak of this branch of the art, it seems, however, that this will be the ultimate solution of the propelling question for marine purposes for many years to come. The absence of reciprocating parts, the saving in weight and space, the absence of vibration, and no doubt considerable increase in economy, make it desirable that investigation should be furthered in this line. But, irrespective of this improvement, for economical reasons the steam engine must soon be supplanted by some other form of generating power. In this

article an endeavor has been made to show what can be accomplished without any very radical departures in any instance from what is being done to-day, by simply taking advantage of the best practice on land and applying the same to marine purposes; a plant of very high economy, very low cost, light weight and small space has been designed and offered for the consideration of those interested.



BRITISH STEAMER INVERIC, AFTER COLLISION.
[Photograph by James Fisher, Hull, Eng.]

The Steamship Inveric in Collision.

The accompanying photograph is of the damaged bows of the steamship *Inveric*, after a collision off the coast of Spain on June 6, 1904, with the steamer *Kate B. Jones* of Cardiff. The latter steamer was very badly damaged, but the captain managed to run her aground, thus saving her from sinking in deep water. The *Inveric* was on a voyage from Calcutta to Hull, and the two ships collided in a dense fog. Immediately after the collision the captain of the *Kate B. Jones* put his wife, niece, and the ship's cook into a boat, but they were never seen again. The damaged bows of the *Inveric* were covered over with canvas and sails and she proceeded to Hull without assistance from any other vessel.

When the vessel was unloaded at Hull tenders were asked for the repairs, and she was sent to a firm at Newcastle-on-Tyne which completed all repairs in fifteen working days. The ship was built in 1901 by I. W. Hamilton & Company of Glasgow, and is owned by A. Weir & Company. The dimensions of the vessel are: 369 feet length between perpendiculars; 52 feet 1 inch molded beam, and 27 feet 2 inches molded depth.

NAVAL ARCHITECTURE.—III.

BY SIR WILLIAM H. WHITE, K.C.B., F.R.S.

During the last ten years no marked advance has been made in our knowledge of the scientific principles on which estimates for the structural strength of ships are based; but very much has been added to available data by the widened application of methods of calculation and comparison, for which we are largely indebted to the late Professor Rankine. In that admirable work, *Shipbuilding, Theoretical and Practical,* to which he made most valuable contributions, Rankine not merely indicated general procedure, but gave illustrative examples of methods of computing bending movements and sheering stresses, and regulating the longitudinal and transverse strength of ships. Mr. William Froude also dealt with this branch of naval architecture in a masterly and suggestive manner. Later investigators have advanced the investigation, particularly in connection with the transverse strength of ships, and the effects of rolling and pitching motion upon stresses borne by ships' structures. The Transactions of the Institution of Naval Architects contain the most valuable contributions to such knowledge. After making allowance for all that has been done in this direction, it still remains true—and, from the nature of the case, will probably always remain true—that the safest rule for naval architects is to base scantlings for new vessels upon comparisons with other vessels with which experience in actual work has been gained. Such comparisons must rest upon thorough analysis of all features in the problem which admit of exact treatment, and due allowance must be made for differences in forms, dimensions, and lading. Careful study of cases of partial failure, exceptional straining or other misfortunes to ships frequently furnish valuable information for guidance in future design, although, to individuals, they may be unpleasant and troublesome. Increase in the speeds and dimensions of ships have introduced many new conditions, and, in some instances, there have been indications of local or general weakness requiring modifications of structures; but, in proportion to the strides that have been made during recent years and the departures from precedent, cases of serious general structural weakness have been neither numerous nor costly. No doubt, we are still imperfectly informed as to the actual margin of strength existing in many structures, and it cannot be asserted that accepted systems of construction embody the best conceivable distribution of material for the association of lightness and strength. Brunel, in a famous passage of his notes on the *Great Eastern*, laid down the true principle of construction when he said that "no materials should be employed on any part except at the places, and in the direction, and in the proportion in which it is required, and can be usefully employed for the strength of the ship; and none merely for the purpose of facilitating the framing and first construction." This broad generalization is often overruled in practice by the desire to secure greater economy or rapidity in building, or by considerations of a practical nature involving accommodation, cargo-carrying power, or facilities for working cargo. In war ships, in particular, considerations of armament and protection often interfere radically with arrangements which, from the point of view of strength of structure, would be preferred. The installation of enormous engine power also introduces local conditions of an important character which cannot be disregarded. After making allowances for all these circumstances, the writer is disposed to believe that Brunel's principle could, with advantage, be more largely applied in ship construction than it has yet been; and the matter is one which must be dealt with courageously if full advantage is to be taken of the superior qualities and forms in which materials for ship building can now be obtained.

IMPROVEMENTS IN MARINE ENGINEERING.

Since 1894, there have been many improvements in the propelling apparatus of ships, tending to economies in weight and coal consumption. It would be out of place to enter into any detail of these, but they so greatly influence ship design that a brief reference must be made to the main lines of advance. From 1891 to 1901, it has been estimated by high authorities that for British

mercantile steamers the average steam pressure rose from about 160 to 200 pounds per square inch; the average revolutions from about 64 to 87 per minute; the average piston speed from about 530 to 655 feet per minute; and the average coal consumption changed from 1.75 to 1.5 pounds per horse-power hour on prolonged sea voyages. With cylindrical boilers (tank type), which are universally used for merchant ships, 220 pounds per square inch is the maximum pressure adopted, and many engineers prefer to use a somewhat lower pressure. Steamers of high speed have higher rates of revolution, the piston speeds on service being 900 to 1,000 feet per minute. There is a consensus of opinion in favor of retaining long strokes in fast ocean steamers.

War ships are engined on a different system, largely because they ordinarily cruise at low speeds and rarely steam at maximum powers. Shorter strokes and higher rates of revolution are adopted, with greatly lessened weights, in proportion to maximum power and large economy of space. For battleships, 100 to 110 revolutions and piston speeds of about 900 feet per minute are used; for large cruisers, 120 to 140 revolutions and 1,000 feet piston speed; for small cruisers, 220 revolutions and 1,000 feet; for destroyers, 350 to 400 revolutions and 1,200 feet.

Water-tube boilers are now universally adopted for all classes of British war ships, with steam pressures of 250 to 300 pounds. The long-continued controversy on this subject has not ended; but enlarged experience has led to great improvements, and with recent types of boilers greater endurance and economy in coal consumption have been secured. The Belleville type has recently given better results, and there are many advocates of its extended use in British war ships. The advisory committee appointed by the admiralty do not favor this policy; and on their recommendation the three types used at present for large ships are the Yarrow (large tube), Babcock and Wilcox, and Niclausse. As a temporary expedient, and to secure economy of coal consumption at cruising speeds, the committee recommended the use in large ships of a small number of tank boilers in association with water-tube boilers. Personally, the writer never favored such a combination. It has been largely used in Germany. As was anticipated, it has been abandoned almost before the first examples were tried, the economy of coal consumption with recent water-tube boilers having proved greater than that of their predecessors.

With water-tube boilers in British war ships open stoke holds have been adopted, in association with powerful fans. In merchant ships some system of forced draft is largely used, that introduced by Mr. Howden finding most favor.

In the smaller classes of cruisers and vessels of the torpedo flotilla, water-tube boilers, with tubes of small diameter, and machinery of the "express" type, are necessarily adopted. The latest and largest application of this system is in the scouts now building. These vessels (360 to 370 feet long and about 2,900 tons displacement) are to have engines of 16,000 to 17,000 I. H. P., driving twin screws. The *Novik,* of the Russian navy, is very similar, but she has three screws. Great interest attaches to the trials of these vessels, particularly in regard to the utilization of their great power and the management and endurance of the engines. In such installations, closed stoke holds and high forced drafts are adopted. Recent trials have shown remarkable economy in coal consumption of destroyers, in association with moderate forced draft. Messrs. Yarrow have obtained, or exceeded, the full contract speed in some instances with less than 2 inches of water as the air pressure in stoke holds, and about 1.6 to 1.7 pounds of coal per horse power hour.

STEAM TURBO-MOTORS.

The practical application of steam turbines to ship propulsion, especially by Mr. Charles Parsons, has been delayed by various accidental circumstances, and partly by the necessity for solving many new problems. Encouragement was given by the admiralty to the new system, immediately after the *Turbinia* had been made successful, by ordering the destroyer *Viper.* This encouragement has been continued, not merely by ordering other destroyers similarly engined, but by extending the system to the *Amethyst,* a third-class cruiser, 360 feet in length, and 3,000 tons, with engines of about 10,000 I. H. P. This was the writer's last design of a small cruiser for the Royal navy, and he recommended the use of turbines. Her trials are now about to be undertaken, and will be of great interest. If they prove successful (of which there can be no doubt) it may be anticipated that the admiralty will apply turbines to much larger vessels.

Meanwhile the greatest experience has been gained with passenger steamers employed on cross-channel or coast service. Messrs. Denny showed the way with the *King Edward* for the Clyde, and the *Queen* for the Dover-Calais route. Now a large number of similar vessels are built, or building. Several yachts have been fitted with turbines. One of the latest of these was designed by the writer for Sir George Newnes, the turbines being specially arranged to give economy of steam at cruising speeds, which will be accompanied by the development of about 50 per cent. of the full power.

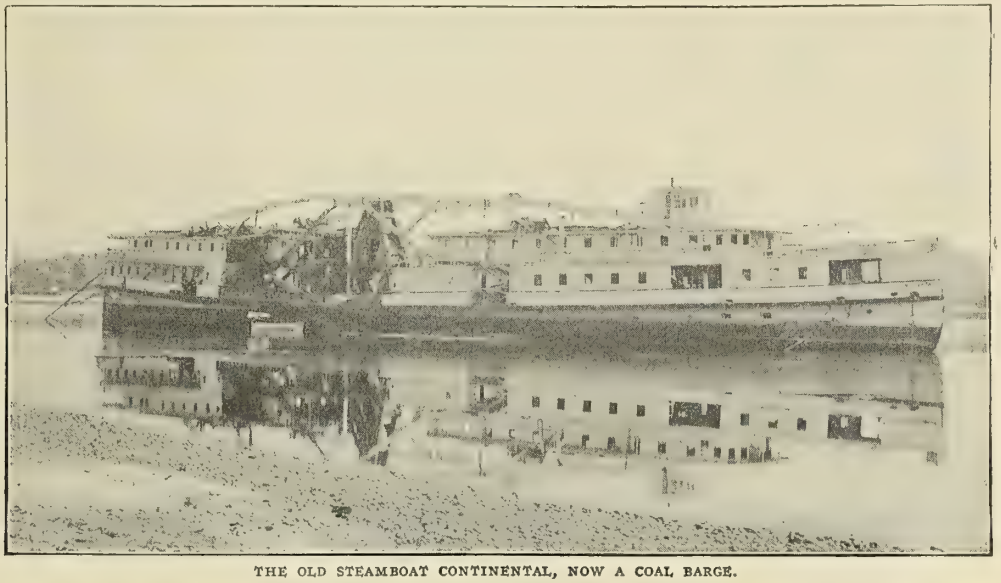
On the whole, experience with turbine machinery has been satisfactory, and its advantages in regard to freedom from vibration, reduced cost of working, maintenance and supervision, and diminished space and weight have been conclusively demonstrated. As to economy in the use of steam as compared with the best types of reciprocating engines, the most trustworthy information has been obtained in electric generating stations on land, where it has been proved that turbines are much superior at full power, about equal at 70 per cent. of the full power, and inferior to reciprocating engines for smaller percentages. Mr. Charles A. Merz, of Newcastle-on-Tyne, has probably made the most complete experiments on this matter, comparing reciprocating engines of high efficiency with turbines. As a result he has adopted turbines exclusively in his latest and largest generating station. Another fact, experimentally demonstrated, is that as the size of turbines increases so their relative economy is improved.

No doubt in destroyers and small vessels fitted with turbines it has been found that at low cruising speed and with small powers —not exceeding 10 per cent. of the maximum—the reciprocating engine has given much greater economy than the turbines actually fitted. This feature of turbine machinery is especially important for war ships, and is receiving close attention from Mr. Parsons and other engineers. Already modified arrangements of turbines and combinations of reciprocating engines and turbines are being tried. One of the most interesting experiments in this direction is that of Messrs. Yarrow, wherein a central reciprocating engine is associated with two turbines, each of the three developing about the same power at full speed, while at cruising speeds the central engine is used for propulsion. In the *Amethyst* this matter has been specially dealt with, and the comparative trials of that vessel alongside of sister cruisers with reciprocating engines will be of great interest and value.

Three or four shafts have been used in most of the turbine steamers, three, as a rule, being preferred. It is well known that, under the new conditions of extremely rapid rotation, the selection of the most suitable propellers has been a matter of great difficulty. In the *Turbinia* this was especially true, and Mr. Parsons showed the greatest courage and ability in the various modifications by which he achieved ultimate success. Very much remains to be done in this direction, and experiments alone can furnish the best solution. Reversing arrangements were not fitted in the *Turbinia,* but are essential to manœuvring power, and are now universally adopted. In some cases it appears that somewhat greater power in the reversing turbines would be advantageous. This is a detail easily dealt with when requirements are clearly stated. Of course such additional reversing power involves greater weight. In the *Turbinia* an extreme example was given of what the system could produce in regard to the proportion of weight to power. Subsequent experience has tended to lessened rates of revolution, increased reversing power, and other modifications tending to increase of weight. It is, however, undoubtedly true that turbines are relatively lighter than reciprocating engines. Messrs. Denny estimate

that if the turbine steamer *King Edward* had reciprocating engines instead of turbines (the same boilers remaining), the speed would be reduced from 20.5 to 19.7 knots, corresponding to 20 per cent. difference in power.

In ocean-going passenger steamers built to work regularly at full power many of the difficulties above mentioned do not occur. The occasions are few, and not of long continuance, where the engines are worked at low power. Consequently the turbine system, for equal speed, should have a two-fold advantage, *viz.*, by reason of the less weight in proportion to power, and the more economic use of steam. On these points actual experience will soon be gained with the two turbine steamers now building for the Allan line. They are 520 feet long between perpendiculars and about 12,000 tons gross. The boilers are capable of supplying steam for 11,000 I. H. P. with reciprocating engines. There are three shafts, each driven by a turbine and carrying a propeller of small diameter. The average speed in moderate weather is to be 17 knots, but a higher speed should be readily obtained. An interesting comparison may be possible between the performances of these vessels and those of the *Moldavia* class of the P. and O line, which are very like in dimensions and power, but fitted with twin-screw reciprocating engines of the most modern type.



THE OLD STEAMBOAT CONTINENTAL, NOW A COAL BARGE.

The decision to adopt the steam turbine system in the new Cunard steamships has aroused much interest, and surprise has been expressed in some quarters that an experiment of such magnitude should have been undertaken. It is, however, well known that before reaching the decision the chairman and directors appointed a strong and representative committee, who thoroughly investigated the matter experimentally and scientifically. The report of that committee has hitherto been treated as confidential by the Cunard Company, and any publication on the subject has been unauthorized and incomplete. As a member of the committee, and as consulting naval architect of the shipbuilding company, who are responsible for the design and construction of one of the vessels, the writer is fully informed, but not at liberty to enter into details. One point, however, may be dealt with as of general interest. Those who consider the course taken to be doubtful, naturally dwell upon the fact that, at one stride, an advance is to be made in the power of the turbines (as compared with the largest engines of that class yet fitted in any ship) fully as great as that which has been made, by successive steps, with reciprocating engines in the last forty-five years. It cannot be questioned that this great stride involves certain difficuties, particularly in the design of details and the conduct of manufacture. Nor is there any doubt but that experience—with its processes of trial and error, and its suggestions for successive improvements—is of immense value, when it is possible. This is the policy recommended by the writer for the Royal navy and adopted by the admiralty for turbine-propelled vessels. But the case of the Cunard steamships was altogether special. In order to obtain the desired speed an increase of at least 70 per cent. of engine power was required over that of the swiftest and most powerful existing Atlantic liner. As a consequence triple screws would have become necessary; and, even then, each engine would have largely exceeded in power any previous twin-screw engine, while there would have been little experience to go upon in designing efficient propellers, and these, with the reciprocating type of engine, would have been of great diameter. Moreover, the installation of three sets of machinery would have involved considerable difficulty, and one set would necessarily have been placed far aft, where the danger of serious vibration, even with the best balanced design possible, would have been great. Altogether it is certain that if reciprocating engines had been used in these vessels many experimental features would have been introduced, and some of these involved serious risks and drawbacks. These considerations are undoubtedly weighty, but they have been overlooked in some criticisms.

One incidental result of the use of quick-running steam turbines has been the extended use of triple and multiple screws, and the enlargement of experimental data in that direction. The writer's opinion in regard to triple screws as compared with twin screws, when reciprocating engines are used, is well known, and need not be restated. But with turbines the conditions are altogether different. Triple screws become a necessity in most cases, and in the Cunarders, quadruple screws were the best solution of the problem, having regard to engine design and propeller efficiency. The small diameters of screws possible with quick-moving turbines give great advantages, including deeper immersion, less probability of racing, and less augment of tow-rope resistance. Naturally experience alone can decide what is the best form and area of blades that can be adopted for each design; and it may well happen that in this particular the first trials may suggest the possibility of improvement in pitch ratio, or in blades. This is quite a common experience now, when unprecedented speeds are attempted, and all that can be done in either case is to take all possible advantage of previous experience in designing new propellers. It need hardly be added that this has not been overlooked in the Cunard designs.

Steamboat Continental.

In the steamboat world, as well as elsewhere, the old must give way to the new, and every day sees the laying up and demolishing of old steamers which have been supplanted by vessels of modern construction. The accompanying photograph was taken of the former Long Island sound steamboat *Continental,* belonging to the New Haven line. She is now a coal barge.

PRACTICAL POINTS ABOUT THE SCREW PROPELLER.—I.

BY W. F. DURAND.

In this and following articles an effort will be made to treat the screw propeller in a simple and practical manner with reference to form and structure, delineation on the drawing board, design and adaptation to specific purposes, relation between conditions of operation and results obtained, and features relating to operation under special conditions. The purpose of these articles is to provide a simple answer, in so far as may be possible, to many of the practical questions which are continuously arising in connection with the design, construction, and operation of screw propellers for marine propulsion.

GEOMETRY OF THE SCREW PROPELLER.

We shall first consider the propeller as a geometrical body, and with reference to the varieties which may appear in geometrical form and proportion.

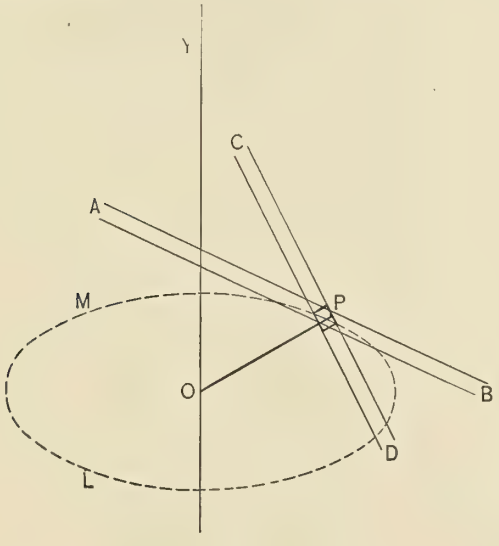


FIG. 1.

In Fig. 1 let *OY* be an upright shaft or axis, and *OP* a radius or line at right angles to *OY*. Suppose now that *OP* is turned about *OY*. The point *P* will trace the dotted circle seen in projection, and the line *OP* will trace the plane of this circle, *PLM*. Then let the rectangle about *P* represent a very small plane element carried on *OP*, and with its plane making an angle *APC* with that of the circle *PLM*. This may be more plainly seen by supposing the element at *P* to be a bit of thin metal with the rod *OP* soldered along one face, the rod being then turned until the piece of metal is tipped up as indicated in the figure. Such an element, if considered very small, would represent in the most general possible way the element of a screw propeller. Let us examine more carefully the relation of such an element to a screw propeller, and how a collection of them would form a propeller complete.

Suppose that while *OP* is revolved about *OY* as an axis it is also moved along *OY* in such way that the element *P* is always moving or sliding edgewise. Then the point *P*, say at the center of the element, will trace a path somewhat as seen in the curve *PGC*, shown in direct vertical projection in Fig. 2. If, again, we had such a curve to start with, it is clear that the element at *P*, Fig. 1, would exactly slide along such a curve, while the rod *OP* carrying it would turn, and at the same time move along the axis *OY*, with motion similar to that previously described.

Such a curve is known as a helix. Again, it is clear that each and every point of the radius *OP*, Fig. 1, will, when it is moved in this manner, trace a similar curve, and that the collection of such curves would constitute a surface. The same surface may also be considered as a collection of all the positions of the radius *OP*, Fig. 1, in moving as above specified. That is, if the axis *OY* were supposed to be drilled full of very small holes very near together, and in each one a rod were placed in such position as to just touch the curve *PGC*, then it is clear that the collection of such small rods or lines would form a surface somewhat as indicated in Fig. 2. It is also clear that such a surface may be considered as made up of a vast number of little elements similar to *P*, Fig. 1, but set at varying angles in accordance with the distance of the element from the axis, as will be seen more fully at a later point. A surface thus formed is known as a helicoid or helicoidal surface, and forms the face of the usual type of uniform pitch propeller with blades at right angles to the axis.

We must now turn back to the helix *PGC*, the path of the point *P* as it turns about *OY* as axis. It will be clear that, as defined, the point *P*, Fig. 1, and hence the curve *PGC*, Fig. 2, lie on the surface of a cylinder with *OY* as axis and *OP* as radius. Such a cylinder is represented in Fig. 2 by *CDFP*, with *OY* as the axis. Suppose now that we imagine the surface of such a cylinder formed by a sheet of paper and this sheet to be split down the element *CP* and then unwrapped. It will unroll or develop into a rectangular sheet shown by *PQRS*, Fig. 3, and the curve *PGC*, Fig. 2, will become a straight line, *PR*, Fig. 3. In Fig. 2 we have shown that part of the curve which would be traced by carrying the radius *OP*, Fig. 1, through one complete revolution about *OY*, or until in Fig. 2 the radius *YC* becomes parallel to the starting position at *OP*. Then, in Fig. 3, the developed curve will form a complete diagonal, *PR* of the rectangle *PQRS*.

The distance *PC*, or *FD*, or *QR* is called the pitch of the helix, or of the helicoidal surface. It represents, as may be seen, the distance which the helicoidal surface would progress forward were it to make one complete turn about its axis and at the same time slide along a smooth supporting surface of the same form. This is exactly realized by the movement of a bolt in a nut, and the ordinary screw thread of the bolt is a form of helicoidal surface similar to that under present consideration. In fact, for a square thread with the flanks at right angles to the surface, the flank of the thread is formed of a part of a helicoidal surface exactly such as that under present discussion.

This gives us the idea of pitch for the helicoidal surface as a whole. Suppose, then, that with such a surface as a support we

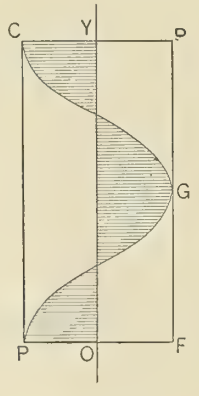


FIG. 2.

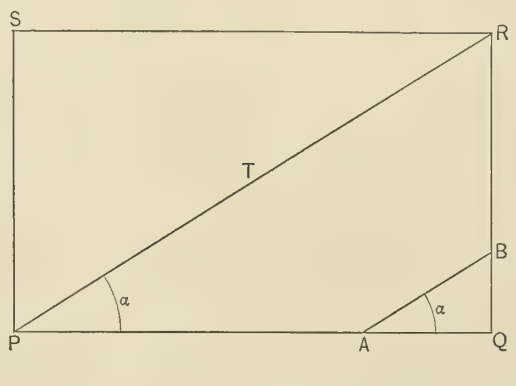


FIG. 3.

again take a single small element as in Fig. 1. Then if this element so moves as to slide along, always supported on the surface of the helicoid, it is clear that together with its radius, *OP*, it will, for one complete revolution, move axially this same distance, *PC* or *QR*, Fig. 2 or 3. This, again, is exactly similar to the movement of a small bit of the thread of a bolt sliding over the thread of a nut. It is important to see in this way that the idea of pitch belongs to the element *P* just as much as to the entire helicoidal surface in Fig. 2. It thus appears that the pitch of such an element as that in Fig. 1 is the distance which would be moved axially for a complete revolution of the radius *OP*, supposing the element to be supported on a corresponding smooth surface along which it can slide continuously, and without difference of direction between element and support at any point of contact between the two.

Let us then consider that we have thus developed a definition of pitch which relates to a small element such as that in Fig. 1. Pitch belongs then essentially to the element, and is determined in accordance with the definition above. This idea is of great importance and should be clearly grasped.

Now, it will be clear from these definitions, that at every point of a surface such as that in Fig. 2, the element has the same pitch, namely, the distance *PC* or *QR*, Fig. 3, and that a propeller made up of blades having a driving face formed of any part of this surface will also be of the same uniform pitch measured by *PC* or *QR*. To form the blade of an ordinary uniform pitch propeller we have only to select a part of such a helicoidal surface for the face, put on thickness so that it may be formed in actual materials, supply a hub, and the blade is ready for operation; and with two or three or four such blades, all similar and symmetrically distributed about the circumference, the propeller is complete.

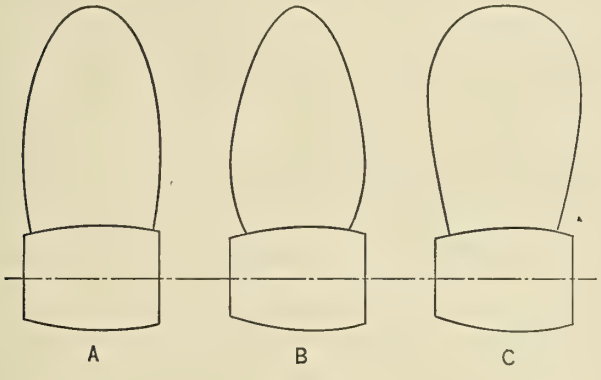


FIG. 4.

The contour of the blade is usually determined with reference to its so-called developed form. Strictly speaking, a surface such as that under consideration cannot be developed or rolled out flat without deforming or stretching it. The helicoid belongs to the same class of surfaces as the sphere in which the surface cannot be rolled out flat without deformation or stretching. In most propeller blades, however, the amount of such deformation would be very small, and we may in a general way safely assume that if a propeller is laid down on the drawing board as though the blade were flat, the actual blade when cast in the foundry will differ in area by no serious amount from that originally intended.

The actual developed shape of propeller blades in most modern cases is oval, or closely elliptical, as represented in Fig. 4A. In some cases the oval is somewhat pared off toward the outer end, as in Fig. 4B, massing the area relatively nearer to the hub; while in others the oval is broadened at the outer end in varying degrees, as indicated in Fig. 4C.

The propeller thus resulting, and as shown in Fig. 5, is of the common type with uniform pitch, and with blades at right angles to the axis. Let us now examine the more important variations from this common type.

Suppose, In Fig. 1, that *OP* instead of being set at right angles to the axis is tipped up or down at an angle as shown in Fig. 6. Then we may imagine *OP* revolved about *OY* with *P* guided on a helix exactly similar to *PGC*, Fig. 2. This movement of the radius *OP* will then generate a helicoidal surface somewhat simi-

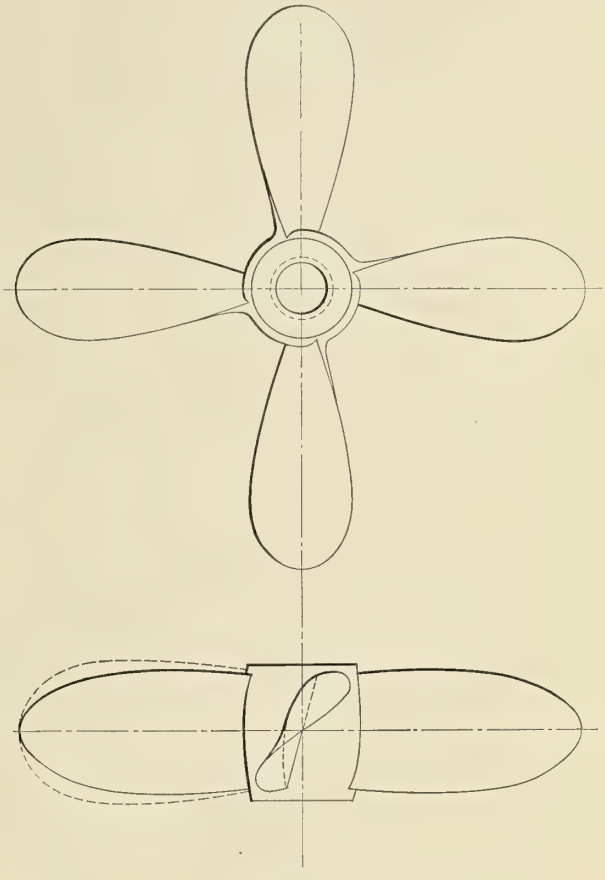


FIG. 5.

lar to that indicated in Fig. 2, but with the straight radial elements inclined at an angle instead of square with the axis. This surface is similar to that which would be formed in cutting a screw thread with a square-nosed tool if it were fed by a com-

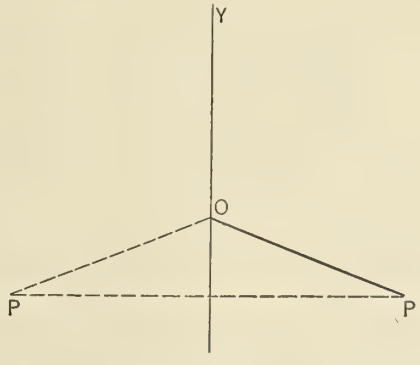


FIG. 6.

pound rest obliquely instead of square with the rod. It is, moreover, exactly the form of surface produced on the sides of a V-shaped thread as cut on an ordinary bolt. Now, the contour of a propeller blade may be laid off on such a surface as readily as on that in Fig. 2, and thus is formed a blade which stands tipped back from the plane of rotation, or with its radial elements oblique to the axis instead of square. As actually employed, the blades are tipped aft rather than forward, thus increasing clearance

between propeller and stern post, or accommodating the location of the tips of the blades better with the arch of the propeller aperture. Such a propeller is shown in Fig. 7.

Again the shape of the blades is by no means symmetrical about a blade center line as indicated in Fig. 4. Quite commonly the leading or forward side is pared away and the after or following side is filled out somewhat as indicated in Fig. 8. Various modifications of this character may be made, and the area may be so distributed as to give to a propeller with elements at right angles with the axis the appearance of being bent or twisted in a variety of ways. As a rule, however, the modification thus made is of the nature shown in Fig. 8, *AB* denoting the after side of the blade.

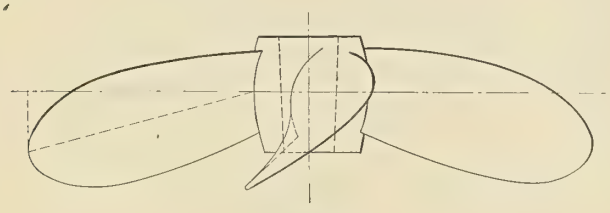


FIG. 7.

Turning again to Fig. 1, it is seen that the pitch of the propeller is determined by the angle between the element *P* and the plane of rotation, or *APC*. This is also equal to the angle *QPR*., Fig. 3, as may be readily seen. Now, the equality of pitch for the helix *PGC*, Fig. 2, at all points means simply that at all points on the curve the inclination is the same, and this is further shown by the development of the helix into a straight line, *PR*, which at all points is inclined at the same angle with *PQ*. Now, for Fig. 3, suppose we substitute Fig. 9. Let *PUR* be a curve, as shown,

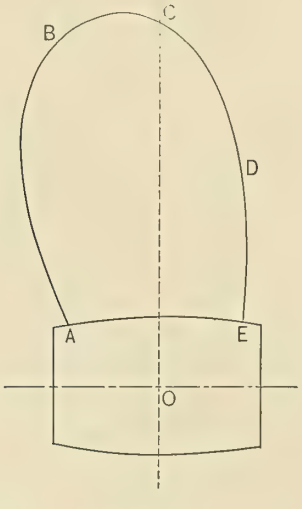


FIG. 8.

and suppose the rectangle wrapped up to form a cylinder. Then the curve *PUR* will form a helix, *PGC*, somewhat as shown. From *P* continuously to *R* the slope of the line *PUR* decreases, and, consequently, the pitch of an element of the helix will continuously decrease from *P* along *PGC*, Fig. 10.

We may obtain further light on this point by remembering that the pitch at any point as *P*, Fig. 9 will be the same as would be determined by the inclination of a line drawn tangent to the curve at this point. That is, it is the same as that of a uniform pitch helix having the same inclination as that at the given point. Such a tangent line is shown at PR_1, and the pitch is QR_1. In general, from the definition of pitch developed in connection with Figs. 2, 3, it will be seen that in any case the pitch of an *element* may be found by the following construction:

(1) Draw a base line such as *PQ*, Fig. 3, and equal in length to the circumference of the circle traced by the element.

(2) Draw at *Q* a line at right angles to *PQ*.

(3) At *P* lay off an angle *QPT*, equal to the pitch angle, and prolong the oblique line thus determined to meet the vertical at *R*. Then the distance *QR* will represent the pitch.

Applying this construction to the present case it is clear that we may find the pitch for any point *S* on the line *PUR* or its corresponding point on the helix *PGC*, by the following construction:

(1) Through the given point *S* draw a line tangent to the curve *PUSR*.

(2) Through *P* draw a line PR_2 parallel to this tangent.

Then the distance QR_2 will denote the pitch for this particular point in the helix or for any element of a propeller having the same radius and inclination. It thus appears that for such a helix as that in Fig. 10 the pitch will vary between limits as shown by *QR*, and QR_3, the latter being determined by a line PR_3 parallel to the tangent to the curve at *R*. Suppose now that as in Fig. 2 a radius revolves about *OY*, Fig. 10, and rests upon or is guided by *PGC*. Such radius may, moreover, be square with the axis, as

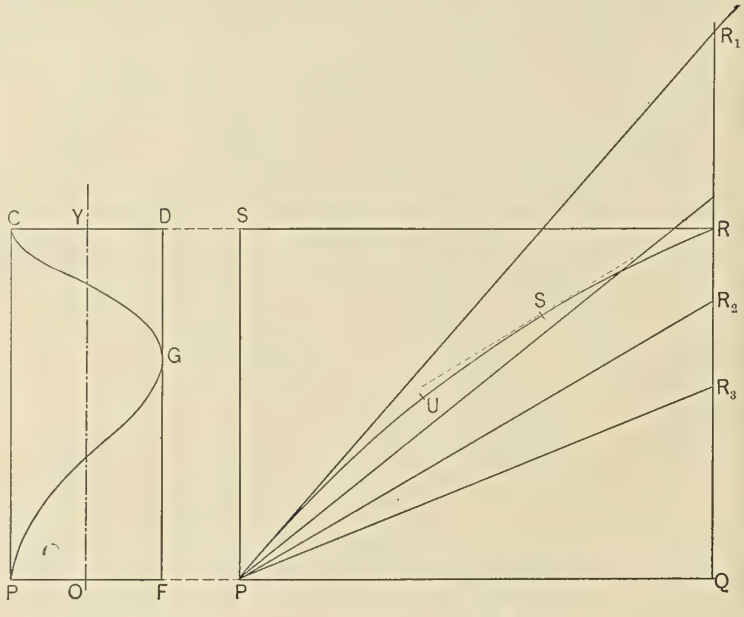


FIG. 10. FIG. 9.

in Fig. 2, or inclined as in Fig. 6. A helicoidal surface will in either case be generated, based on the helix *PGC* as guide, and with the same variation of pitch as the helix, and as indicated in Fig. 9. Now, if a portion of such a helicoidal surface were taken for a propeller face, it is clear that the pitch would continuously increase across the face in the direction from *R* toward *P*, and decrease in the opposite direction. When the pitch varies across the face in this manner, it is made to increase from forward aft or from leading to following edge. In such case, therefore, if the part *US* should represent the portion of the curve actually embodied in the blade, the part toward *S* would give the leading edge and that toward *U* the following edge.

It will be clear from the foregoing that any variation of pitch across the face, no matter how complex, may be obtained by an appropriate developed form of helix. Thus, as in Fig. 11, we might have a straight line *PU* running into a reverse curve *USR*, and denoting from *P* to *R* first uniform pitch from *P* to *U*, then decreasing pitch to a minimum at about *S* and then increasing continuously to *R*. Such extreme variations have no especial value with reference to the screw propeller, but serve to show how easily the most complex kinds of pitch variation may be realized if desired.

We have thus far supposed the generating element of the helicoidal surface, as OP, Fig. 2, to be a straight line. This is by no means necessary, and it may be a line curved in various ways, either in the plane of rotation or in the plane of the axis. The former is indicated in plan view in Fig. 12, and the latter in Fig. 13. Such generating lines may be used with a guiding helix having any law of pitch, either uniform or variable, and the resulting propeller will have blades bent or twisted according to the generating line, and pitch according to the law of the guiding helix or that part of it actually used for the blade.

We must now note briefly another variety of pitch variation. It may be seen that if the construction in Fig. 3 for a uniform pitch propeller is repeated for a number of different diameters, a diagram like Fig. 14 will result, in which PQ_1, PQ_2, etc., represent

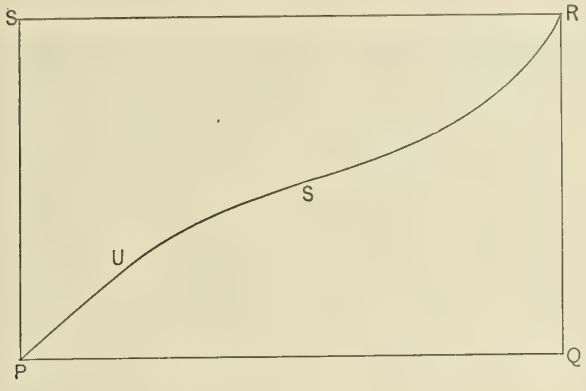


FIG. 11.

the various values of the circumference, and PS, Q_1R_1, etc., represent the constant value of the pitch. On the other hand, suppose that we start with a diagram, such as Fig. 14, taken as a collection of guide lines, and let each one be rolled up into a helix in the same manner as in Figs. 3 and 2. Such a collection of helical curves would then all fit the one helicoidal surface, and each would correspond to the fixed value of the pitch at its own radial distance from the axis. Suppose, however, that instead of PR_1, PR_2, and PR_3, we should take respectively $PR^1{}_1$, $PR^1{}_2$, and $PR^1{}_5$, each with its corresponding helix, for the guiding curve for the part of the surface at the corresponding radius. That is,

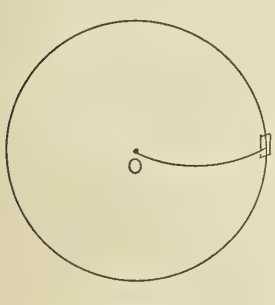


FIG. 12.

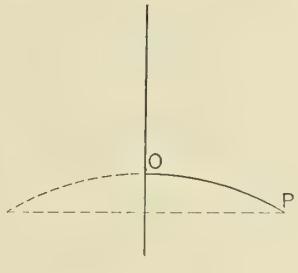


FIG. 13.

we may suppose that we have a surface, helicoidal in character, and yet so formed that the helix resulting from wrapping up $PVR^1{}_1Q_1$ will just fit it at the radius corresponding to circumference PQ_1, the helix resulting from $PTR^1{}_2Q_2$ at the radius corresponding to circumference PQ_2, and similarly for PR_3, PR_4, and $PR^1{}_5$. Then it will follow that at these distances from the axis the pitch will be represented by $Q_1R^1{}_1$, $Q_2R^1{}_2$, Q_3R_3, Q_4R_4, $Q_5R^1{}_5$, and will therefore vary from one location to the next from root of blade to tip.

In a more general way we may represent such a change of pitch by the diagram of Fig. 15. Let PQ_1 represent the circumference at the hub and PQ_5 that at the tip. Then let $R_1R_2R_3R_4R_5$ be a curve giving proportionately the change of pitch from root of blade to tip. Then any line such as Q_2R_2 will give the pitch at the radius corresponding to the circumference measured by PQ_2, the angle R_2PQ_2 will give the inclination of the element, and if the rectangle PQ_2R_2S be wrapped up to form a cylinder, the diagonal PR_2 will form the helix which would exactly fit such a helicoidal surface at this radius. It must be noted that such a surface cannot be generated by sweeping up a single radius in the manner suggested in Fig. 2. In a surface swept as in Fig. 2 the pitch at all points on the radius for any one position is the same while in the surface under present discussion the pitch along any line drawn from hub to tip will vary continuously according to the

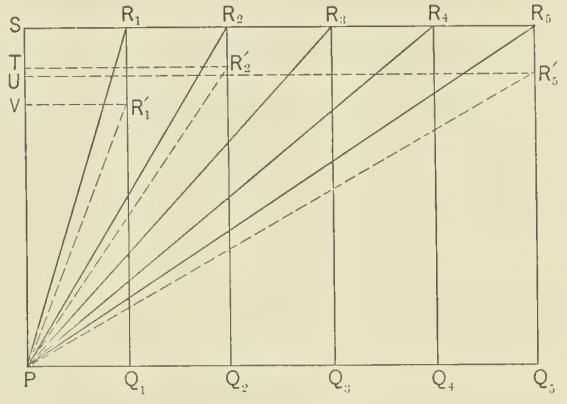


FIG. 14.

law indicated in Fig. 15. Instead, therefore, of trying to consider such a surface as formed of a series of radial elements, as in

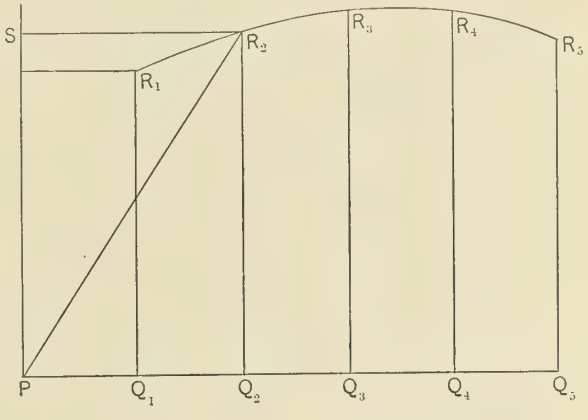


FIG. 15.

Fig. 2, we may consider it as built up of an indefinitely great number of helical curves placed one outside the other, and each with pitch as indicated by the diagram in Fig. 15. In such a propeller, therefore, the pitch would increase from a value Q_1R_1 at the hub to a maximum Q_3R_3 about two-thirds the way toward the tip, and would then decrease to a value Q_5R_5 at the tip.

Variation of the pitch in this manner may also be combined with variation from leading to following edge, as shown in Fig. 9, and in this way the most complex variations in pitch may be produced. The principles involved in the determination or delineation of propellers of any variation of pitch are, however, all contained in the preceding paragraphs, and their application with illustration of further points, will be found in the following chapter on the drawing-board delineation of propellers, both of the standard type with uniform pitch, or with such variations in its distribution over the face as may be desired.

(To be continued.)

ELECTRIC STEERING CONTROL OF SHIPS.

One of the most interesting applications of electricity on shipboard is certainly to be found in electric steering control, permitting the pilot to transmit instantly and in an absolutely sure and certain manner the most definite orders to the steersman, and to receive in return a definite and permanent indication of the position of the helm.

The Allegemeine Electricitäts Actien Gesellschaft, of Berlin, was one of the first companies to elaborate a general design for electric steering control, and the German government shortly after authorized a trial of the same on the naval ship *Baden*. The results having been definite and satisfactory, several armored ships

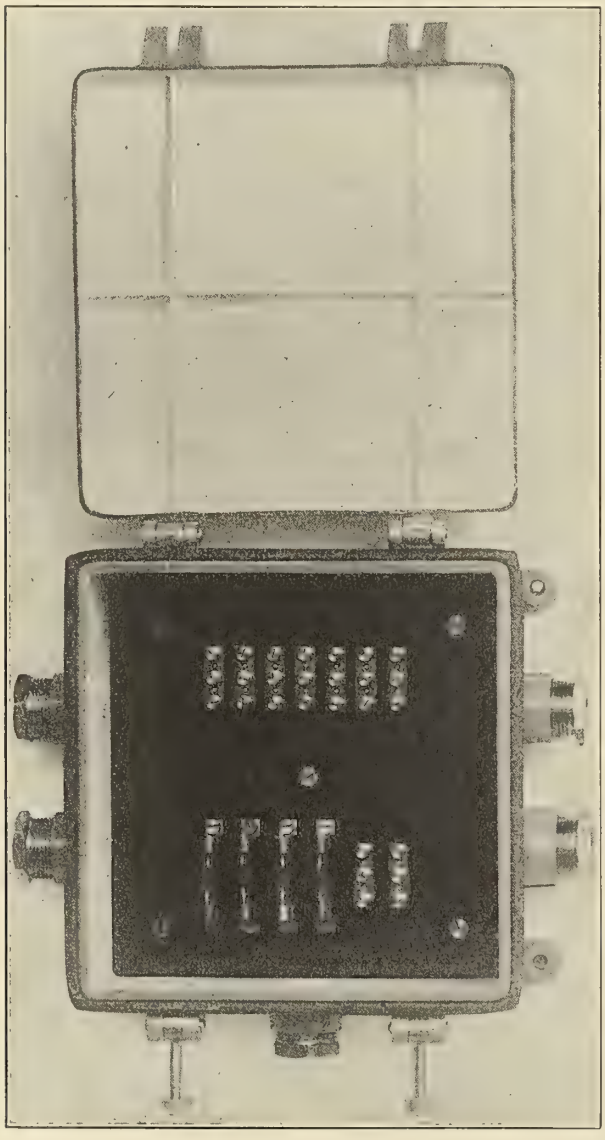


FIG. 1.—BOX I.

were later provided with a similar system, and from indications but a short time will elapse before all the principal units of the German navy are provided with this mode of electric steering control.

The installation on the *Baden* divides itself in reality into two distinct parts, one for the transmission of orders from different command points to the steersman at the wheel, the other for the constant, automatic indication at several points simultaneously, of the position of the helm. These installations comprise the following elements:

1. For the transmission of orders:
   - (*a*) A transmitter placed in the forward conning tower or on the forward bridge.
   - (*b*) A similar apparatus at the central command point.
   - (*c*) A receiver provided with a call bell, and located at the wheel.
2. For indicating the position of the helm:
   - (*a*) A transmitter contact indicator operated by the rudder head.
   - (*b*) An indicator placed in the steering room, in each engine room, on the forward bridge and at the central command point where this apparatus is joined to the two transmitters.

The apparatus is installed in three boxes made carefully watertight; see Figs. 1 and 2, with arrangement plan, Fig. 3. The wiring is installed throughout of marine cable in armored conduit. The current of about 8 amperes for the entire installation is taken

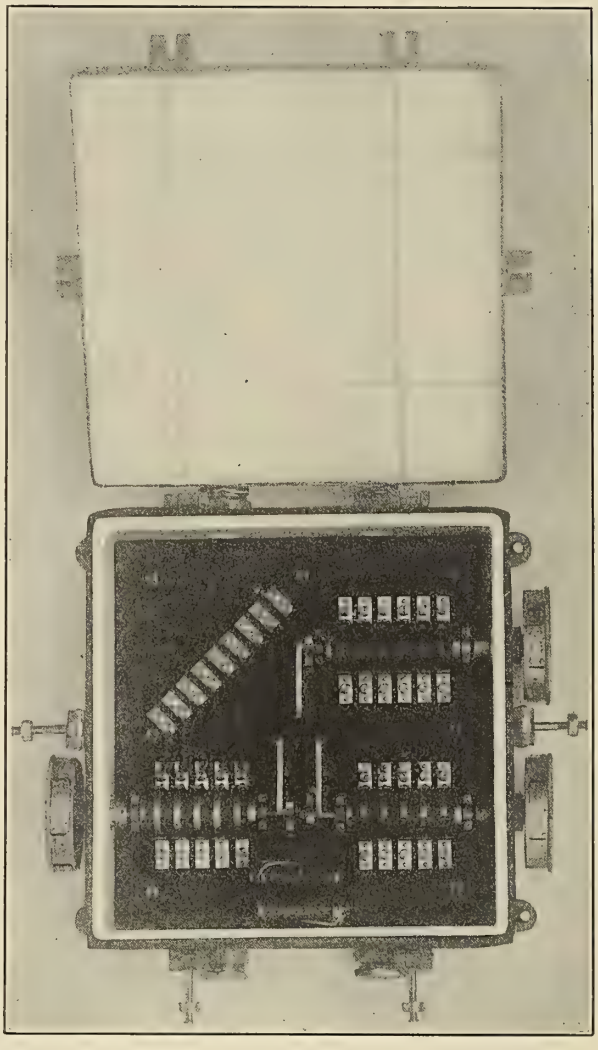


FIG. 2.—BOX III.

from the main lighting circuits by means of a bi-polar commutator with conductors of ample section and leading to a resistance, where the pressure of 67 volts is reduced to 28, the pressure adopted for the apparatus. From this point the current is brought by a cable of five conductors to the distribution boxes. In Box 1, Fig. 1, are gathered the cables of the receiver for the helm telegraph, for the transmitter position indicator of the latter, and for a further indicator. In box 2 are joined the cables for the two indicators in the engine rooms. Box 3, Fig. 2, located at the central point of command, constitutes the point whence are led the cables for the two helm telegraphs, and for the position indicator on the forward bridge. Box 3 is provided with means for cutting out, in case of battle, the double connection from command points, in order to avoid the danger of transmission from different points of orders at the same time.

The cover of this box, Fig. 3, is furnished with reference points, and there are hand wheels with similar marks, thus providing for a ready indication of the connections which may exist at any given moment. The wheel *b* serves to put in circuit the helm position indicator in the armored conning tower, and the wheel *c* that located on the forward bridge. Fig. 3 gives the arrangement plan of the system as well as the interior operation of the apparatus. In order to facilitate the assemblage of the apparatus the cables and connections are marked with reference numbers, as per plan.

Before giving further description of the apparatus employed, brief reference should be made to the principle of the rotating

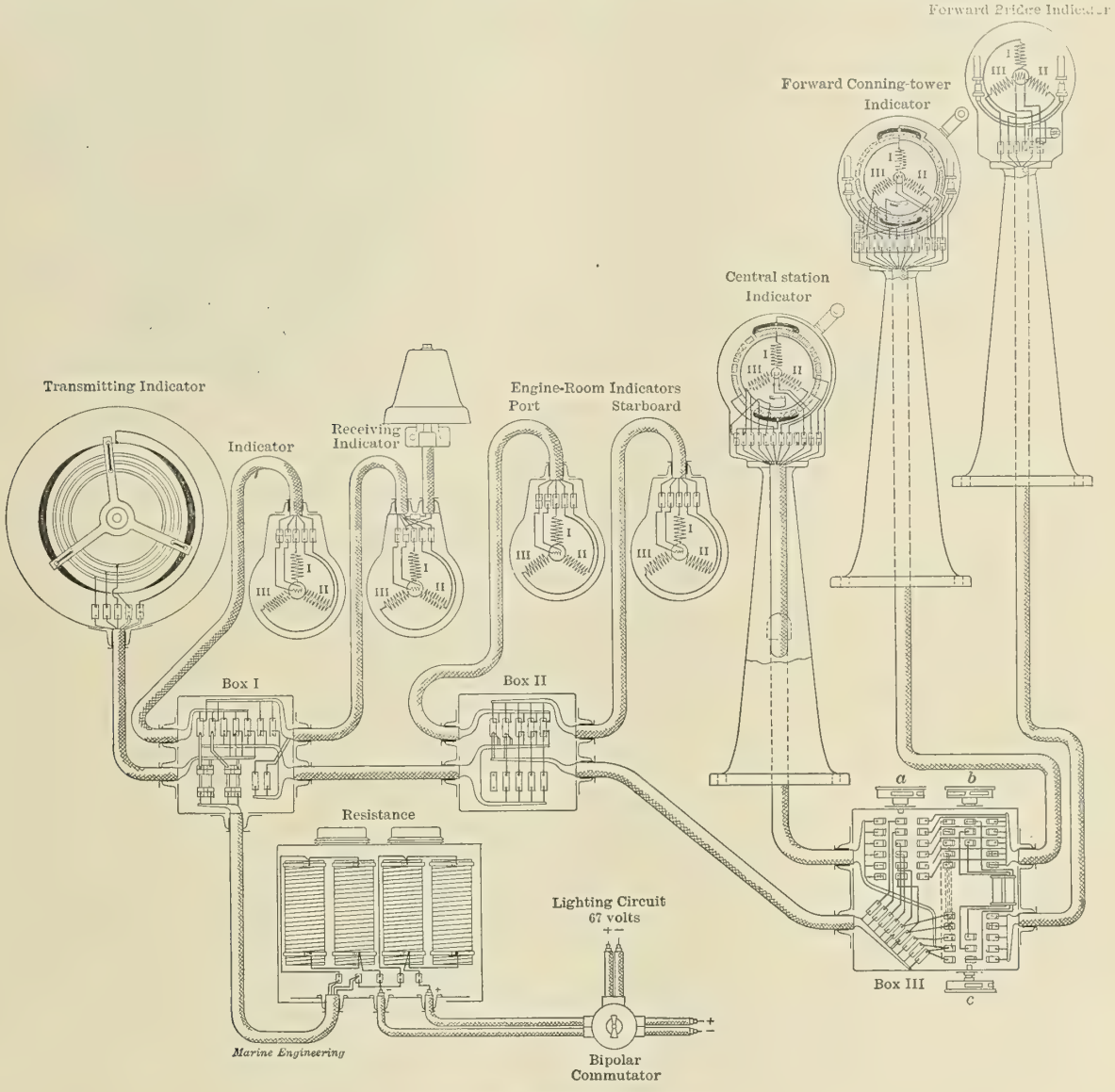


FIG. 3.—HELM INDICATOR WIRING PLAN.

field employed in the position indicators. This system involves the distribution on three conductors of a continuous current by means of a distributor in such a manner that in the receiver, which comprises a tri-polar magnetic system, a magnetic field is constantly developed, electrically dependent on the distributor, with which it rotates constantly in step.

In further description, the term "transmitter" will refer to that part of the apparatus in which the subdivision of the current is effected; the term "distributor" to the rotating part of the transmitter and the term "receiver" to the part containing the resultant magnetic field and rotating magnet.

The transmitter is usually constructed of two different types, for a large or a small number of orders. The transmitter for a large number of orders, Figs. 4 and 5, comprise a certain number of metal blocks and plugs for closing circuit, between which are connected resistance coils, unequal and graduated according to a certain law. This circle of resistances, closed by itself, receives at two fixed points, *a* and *b*, directly opposite each other, the current from the supply point as noted above, while three rotating contact bars, *c, d, e,* at angles of 120 degrees and insulated from each other, take each a part of the current and send it by corresponding conductors to the receiver.

The transmitter for a small number of orders, Fig. 6, consists essentially of three principal contact points with three transmission wires, and a certain number of auxiliary contacts, *b* and *c*, arranged in a circle and connected by resistances I, II, III, with their principal contact point. The current is led to the transmitter by means of two brushes at opposite ends of a diameter and connected with the source of current. The angle of the principal contacts is 60 degrees, and that of the secondary depends on the number of orders for which provision is to be made. A transmitter for 48 orders, for example, requires only 21 contacts. The current is distributed to the connecting ring by two double brushes *d*, during the rotation of which the three wires, G_1, G_2, G_3, joined to the three principal contacts, are connected with varying resistance, the conductors themselves being in multiple, in series or singly, in manner to produce in the magnetic system of the receiver a magnetic field rotating with the distributor, as already explained in connection with the other type of transmitter.

The receiver, see Fig. 7, is composed of three pairs of coils, *r*, disposed radially about a copper cylinder *k*. These coils are in all cases placed in circuit two and two, and connected with one of the transmission lines coming from the transmitter. The ends of the wires from these three pairs of coils are joined to a single terminal where they have their neutral point. At the center of these radiating coils is placed the inductor element, to which is directly attached the indicating needle. The inductor element is formed of two pieces of soft iron, *g*, in the form of a Z, and connected to each other by a piece of insulating material, *h*. The energization of the rotating points, *g*, is effected by means of two coils, *i*, connected up in such manner that there is formed in each Z-shaped core a pronounced north and south pole. The copper cylinder, *k*, surrounding these Z-formed pieces serves effectually to dampen the indicating system.

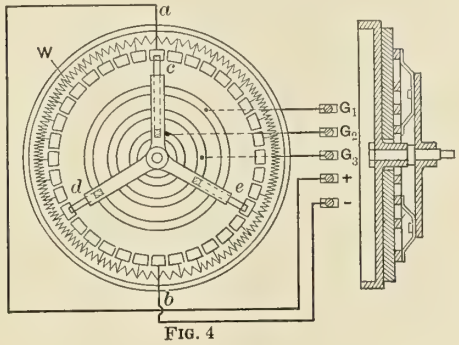


FIG. 4

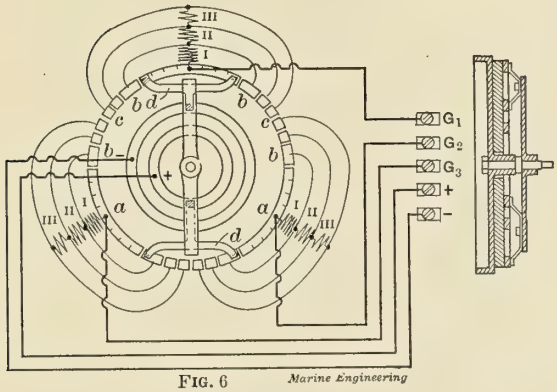


FIG. 6

FIGS. 4 AND 6.

The operation of this type of indicator is as follows: If, for example, the three-armed distributor of the transmitter *I* is moved over the surface of the resistance circle the proportions of the current in three conducting lines are modified according to the resistances inserted; and in consequence of the mutual relations of the three conductors and the resistances involved, the resulting current is sinusoidal in form. As the trace of the current shows, the maxima of current intensity in the three conductors are distributed at relative phases of 120 degrees. These three conductors leading to the three pairs of coils in the receiver, there is produced at this point, and according to the intensity of the indivdual currents, individual magnetic fields which unite in a resultant field, of which the direction always coincides with that of the distributor. This field traverses the central space of the receiver system and forces the rotating inductor element to follow its rotation.

Due to the fact that with this system of transmission one can obtain an unlimited number of positions, it is unnecessary to introduce any gearing connection between the inductor element and the needle moving on the face of the indicator. From these various dispositions it results as follows:

(1) To each position of the lever of the transmitter there can correspond but one position of the needle of the indicator.

(2) The inductor system of the receiver is not reversible, and in consequence any departure of the indicator from synchronism is rendered impossible.

(3) Variations of potential are without effect on the position of the indicator.

As we have already noted, this form of shunt transmitter is

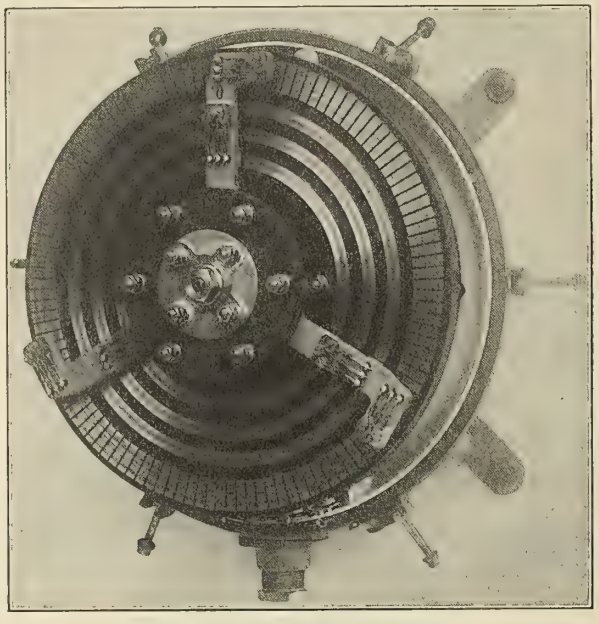


FIG. 5.—TRANSMISSION INDICATOR.

adapted to the transmission of a large number of orders, and is therefore used for the automatic helm position indicator. For the transmission of orders to the helmsman, a much smaller number of indications is required, and the other system is employed.

We shall now refer briefly to the chief constructive details of these two forms of apparatus. For the helm position indicator the general arrangement is shown in Fig. 4. On the anterior face of a disk contact points to the number of 120 are arranged in a circle, together with three collecting rings, while the resistance coils are placed at the back. The three-arm contact system is also mounted on an insulated disk carried on a movable axis. This axis carries

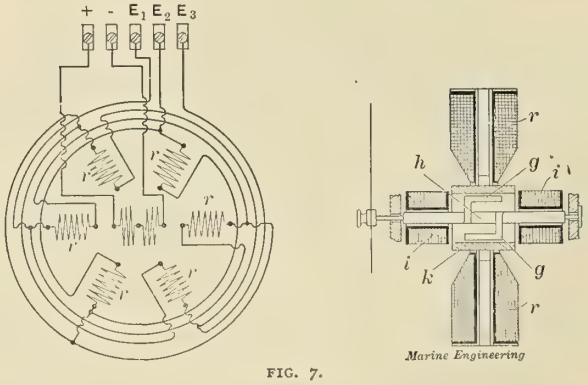


FIG. 7.

on its anterior part a little boss with two grooves, while a similar boss is fixed on the rudder head, see Fig. 8, and thus transmits by means of a steel wire the movements of the rudder head to the transmission system in the direct ratio of 1 to 3. The second groove of the small boss receives the wire of the return mechanism, consisting of a counterweight and tension key.

As indicated by the diagram, the counterweight acts in a direction opposite to that of the movement of the rudder, so that the transmitter is put into operation in one case directly by the pull of the wire from the rudder head segment and in the other by that of the counterweight. The key serves to adjust the tension as may be desired.

For the transmission of orders to the helmsman the general arrangement is shown in Fig. 6, comprising a transmitter for a moderate number of orders, and a receiver. Both transmitter and receiver are mounted in water-tight boxes, from which, however, they may be readily withdrawn without breaking the electric connections. Orders may be transmitted for every five degrees, and for each new order a warning bell signal is sounded automatically at the steering station.

The principles governing the operation of this system of transmission and indication had been known and employed to some extent before its perfection and wider development were undertaken

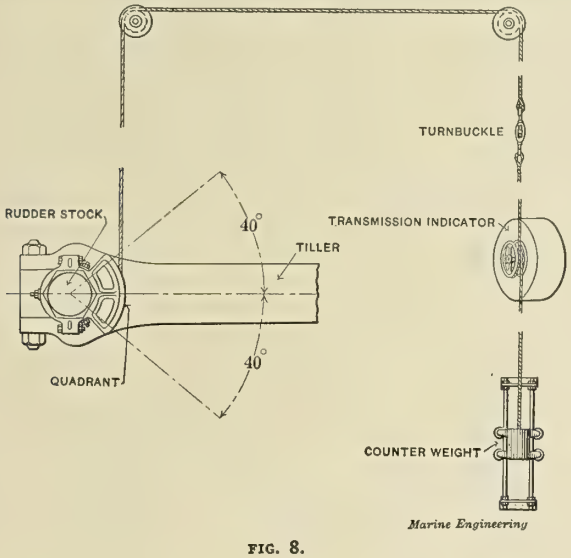


FIG. 8.

by the company referred to above. The chief difficulties have arisen in connection with the protection of the apparatus from the effects of exposure to salt humidity. Oxidation naturally results and the consequences are either the refusal to work or faulty indications. The company whose apparatus is herein described has given special attention to these matters, and by means of stuffing boxes and other points of special construction has developed the structural details in such manner as to insure immunity from disturbances of this character.

Low Bids for War Ships.—Unprecedentedly low bids were offered December 15, for the construction of the battleship *New Hampshire* and the armored cruisers *Montana* and *North Carolina,* proposals for which were opened at the Navy Department at noon in the presence of representatives of shipbuilding companies from the Pacific and Atlantic coasts. The lowest bids offered were for the cruisers, $3,575,000 each by the Newport News Shipbuilding Company, and for the battleship $3,748,000, by the New York Shipbuilding Company. Not more than two ships can be awarded to one company. The lowest aggregated bids for these three ships was exactly $1,377,000 less than the aggregate lowest bids for three sister ships a little over a year ago, a reduction of 11 per cent. in price.

Twenty-Thousand-Ton Battleships.—A 20,000-ton battleship is proposed by some of the government navy experts, as it is claimed it would possess very great offensive and defensive powers. It is proposed to make the main battery uniform and to consist of ten 12-inch guns placed in turret and broadside, while the secondary battery is to be made up entirely of about twenty 3-inch guns. The armor of this type would be but little heavier than that of the *Connecticut* class. The advocates of this radical change point to naval engagements of the Russian-Japanese war to show that the 6-inch gun, which is very largely used on the American battleships, is practically useless against armored vessels at long range, and that the success of the Japanese resulted from the preponderance of heavy guns on their ships. It is also claimed that to resist torpedo attack the new 3-inch gun is as effective as any other.

POWERING SHIPS.—I.

BY EDWIN CERIO, NAVAL ARCHITECT.

I.

One of the principal factors involved in the design of a ship is the indicated horse power necessary to drive her at a given speed. Too much accuracy can scarcely be obtained on this side of the complicated problem, as the elements upon which the solution depends are of such a nature that they cannot, strictly speaking, undergo mathematical investigation. What is aimed at is the best obtainable approximation, and only very accurate and patient work will secure this.

The practice of some designers has often been that of overpowering ships, and thus, by making a sure error in the right direction avoiding a probable error in the wrong direction. But if the simple fact that an error, either way, is always an error, is not enough to convince such designers that their practice is not to be recommended, they must be convinced of this if they look at the economical side of the question, especially in these days of industrial war in which many a battle is fought on horse powers, as the builder who will drive a ship at a given speed with the right amount of power, is sure to offer a cheaper tender than the one who makes the error in the right direction. Of course, the problem of powering a ship is but the determination of the ship's resistance. Resistance of floating vessels has been the object of investigation for many centuries, and to determine it, formulæ of every description have been given, some based on purely analytical speculation, others on experiments, and most of them starting from a completely false conception of the phenomena with which they deal.

It was really only in 1870 that a new era began for such studies, and the problem treated with mathematical and experimental investigation, both very happily combined. The traditional fundamental ideas have been replaced by modern views; an important factor, skin-resistance, has been thoroughly studied and the dynamical law of similitude applied to the theory of ships, so that the problem of resistance of vessels now assumes altogether a new aspect. This is due principally to the classical experiments of Froude, and after him to such men as have helped to popularize his theory—R. E. Froude, Reed, Denny, Risbec, Rota, Durand, Mittendorf, Rauchfuss, Gareis—only to mention a few, and whose works may very profitably be consulted.

The determination of the resistance of ships is done to-day by means of model experiments, using directly for this purpose experimental tanks, or the results gathered by such researches. The first tank was built by W. Froude at Chelston Cross. Froude's example was soon followed in Italy and in France, where Risbec carried out a series of very important investigations at Brest. In Germany, notwithstanding the very little sympathy model experiments enjoy, a tank was built by the North German Lloyd, and another one has just been completed at Charlottenburg, near Berlin. Since the year 1887, in which the tank in Spezia, Italy, was completed, the results of experiments carried out by Rota have added most precious material to the theory of resistance. The United States seem to have realized how such investigations are indispensable, and must be done at home by a nation whose navy has developed so rapidly and successfully. Experimental tanks are also in Russia, and it may be said in all countries in which naval problems are studied and solved. The purpose of these tanks is not only, as is thought by many, to determine for a given ship the power which will drive her at a desired speed, but to gather rationally all the results obtained and to investigate all the questions that arise on propulsion, stability of ships, etc., and it is clear that when these experimental methods become more popular they will furnish naval architects more reliable data than the empirical formulæ which have led, in many cases, to unpleasant experiences.

II. INDICATED HORSE POWER AND EFFECTIVE HORSE POWER.

If V be the speed of the ship in knots, v be the speed of the ship in feet per second, W the resistance in pounds, the effective horse power necessary to drive the ship at the speed V will be:

$$EHP = \frac{W \times v}{540}$$

The indicated horse power is equal to the effective horse power divided by a coefficient of efficiency:

$$E = \frac{EHP}{IHP}$$

This coefficient E depends upon the efficiency of engines and propeller and from many elements which cannot be considered for every single case; therefore, no fixed rule can be given for selecting a value of E suitable to any particular ship. The designer must be guided in this choice by experience and his own judgment; the values given in the following tables will, in all ordinary cases, lead to sufficiently accurate results. Where extreme exactness is not required, for example, when making estimates, the *EHP* may always be assumed to be the half of the *IHP*, and therefore $E = 0.50$.

In Table I. values of E are given for different types of ships.

TABLE I.

	Values of E.
Single-screw merchant ships, full forms, coefficient of fineness between 0.8 and 0.7	0.40
Single-screw merchant ships, coefficient of fineness about 0.6	0.50
Double-screw ships, vertical engines	0.50-0.60
" " " horizontal engines	0.40-0.45
Vessels of very fine forms, great speed, vertical engines	0.55-0.65

The elements upon which E depends are of too complicated a nature to allow any law being given to determine in every special case appropriate values, and the best way to proceed is to get reliable data of ships similar to the projected ship. A few examples may show how the values of E vary, sometimes governed by no progressive law, and sometimes in a certain relation with the speed. The following are the results of trials of:

A TUG BOAT.

Speed in knots	7	8	9	10	11
Values in E	0.51	0.54	0.533	0.535	0.46

A TORPEDO-BOAT DESTROYER.

Speed in knots	12	14	16	18	20	22	24	26	28
Values of E	0.457	0.507	0.533	0.574	0.606	0.618	0.628	0.644	0.666

SOME ITALIAN CRUISERS.

Pellicano.					
Speed in knots	13.13	15.80	17.72	20.34	23.05
Values of E	0.444	0.526	0.478	0.482	0.495
Calabria.					
Speed in knots	12	16.4	14.5		
Values of E	0.4	0.41	0.42		
Dogali.					
Speed in knots	15.5	18.5			
Values of E	0.39	0.45			
Lombardia.					
Speed in knots	10	18			
Values of E	0.48	0.42			
Piemonte.					
Speed in knots	20.17	20.3	22.4		
Values of E	0.41	0.41	0.59		
Agordat.					
Speed in knots	18.78	22.7			
Values of E	0.33	0.417			

When it is necessary to obtain accurate results, it will be well to analyze separately the various causes which influence the coefficient E. All agree in the definition of indicated horse power, which is the mechanical work developed by steam in the cylinders of an engine, taking the horse power as unit; but some confusion still exists in the definition of effective horse power; if the *IHP* is the power required to give the total thrust, the *EHP* is that part of it which is opposed to the net resistance of the ship. Thus defined, as it usually is, the *EHP* appears only as an abstract conception; the relation between *EHP* and *IHP* is of a complicated nature and not easy to be determined. To start from a simple case, we can consider a ship which is being towed; the amount of mechanical work in every unit of time is here equal to the *EHP*. But when propulsion is obtained by other means than towing, the resistance of the propellers, which are attached to the hull, must also be considered, and instead of an amount *EHP* of power, a certain amount $(1 + K)$ *EHP* will be required to obtain the propulsion at the same speed. This amount of power is not all furnished by the propeller, because the vessel is surrounded by a mass of liquid in motion, which may, so to speak, help the propulsion if the stream lines move in the direction the vessel moves in, and therefore the propellers shall furnish an amount of power which may be expressed by:

$$P_3 = (1 - h)\ (1 + K)\ EHP \text{ or } P_3 = \frac{I}{E_3} EHP.$$

The coefficient E_3 depends upon the so-called *ship-performance*, but it must be remembered that it is dependent upon two distinct factors, one $(1 + K)$ which considers the additional resistance of propellers, and the other $(1 - h)$ which depends upon the stream lines and consequently upon the ship's form.

The next co-efficient to be considered is that which takes into account the mechanical performance of propellers, and this is:

$$E_2 = \frac{P_3}{EHP}.$$

If P_2 is the amount of power which is to be furnished to the propeller, that is, the power measured on the propeller shaft, it is necessary that the motive fluid, steam in steam engines, should develop an amount of power, *IHP*, equivalent to the work of the total thrust, or, in other words, equal to P_2 augmented of the mechanical work of all the internal resistances of the engines; the coefficient

$$E_1 = \frac{P_2}{IHP}$$

gives the performance of the engines. The product, $E_1\ E_2\ E_3 = E$, is the coefficient of total performance; as the product $E_1\ E_2$ is independent of the performance of engines, and may be called the net coefficient of performance. The propeller coefficient E_2 has a mean value of 0.65 when the propeller is designed to work in the most favorable conditions of performance; the extreme limits of E_2 are 0.6 and 0.7.

Investigations on the values of E_3 are scarce, but those carried out by means of model experiments seem to prove that in the greatest number of cases, for vessels of ordinary forms, $E_3 = 1$.

In practice it is advisable in considering E_1 and E_2 to refer to their product, $E_1\ E_2$, to which we shall refer, from hence, as E_{I}; the coefficient of performance of the engines we shall call E_{II}. As said above, some confusion exists in the meaning of *EHP*, and especially of that part which depends upon E^{I}; it is measured sometimes abaft, sometimes forward of the thrust block, sometimes at the stern post. To avoid this confusion it would be wise to adopt Herr Föttinger's suggestion* and measure the engine's effective horse power always at the stern tube.

Whenever model experiments for the designed ship may be carried out, and values of E_{II} determined for the propeller and ship, it will be always better to separate E_{I} and E_{II}, because reliable values of E_{I} are always obtainable and may be assumed with sufficient exactness.

According to Riehn:

For big engines $E_1 = 0.80$ to 0.90
small " $E_1 = 0.75$
big ships $E_2 = 0.75$
small " $E_2 = 0.55$.

For most cases the following table may furnish good values of E_{I}

*Fottinger.—Effektive Maschinenleistung und effektives Drehmoment. Yahrbuch der Schiffbau technischen Gesellschaft, 1903.

TABLE II.

E.H.P.	E_I	E.H.P.	E_I	E.H.P.	E_I	E.H.P.	E_I
10	0.575	500	0.68	1,000	0.747	3,000	0.87
100	0.61	600	0.695	1,300	0.775	4,000	0.895
200	0.63	700	0.71	1,600	0.80	5,000	0.90
300	0.65	800	0.72	1,900	0.825	6 000	0.91
400	0.67	900	0.735	2,200	0.84	7,000	0.915

Föttinger investigates separately the different losses of power from the cylinders to the propeller, and mentions the loss due to vibrations, which is usually not taken into consideration.

The following are average values of these losses in percentage of the total horse power developed in the cylinders:

	Per cent.
Loss due to the friction of pistons	0.5
" " " " " " stuffing boxes	0.5
" " " " " " guide blocks	4.0
" " " " " " connecting rod gudgeons	1.5
" " " " " " crank shafts	2.0
" " " " " " bearings	2.0
" " " " " " valve gear	5.0
" " " " vibrations caused by the engines	4.0
" " " " friction of thrust block	2.0
" " " " first shaft bearing	0.4
" " " " second "	0.4
" " " " third "	0.4
" " " lignum-vitæ bearings	2.5
" " " stern tube stuffing box	0.8
" " " the vibrations of propeller	4.0
	30.0

Therefore, of the original *IHP* only 70 per cent. arrives at the propeller, thus giving a coefficient $E_I = 0.70$.

This is a good average value, and as a mean value of E_{II} may be assumed equal to 0.65; when estimating the *IHP* it will always be safe to assume:

$$E = E_I E_{II} = E_1 E_2 E_3 = 0.45.$$

It is scarcely necessary to call attention to the fact that the *IHP* as considered up to this point is only what is strictly indispensable for the propulsion of the ship; a good practice is that of adding to this part of the *IHP* that which is absorbed by auxiliary machinery driven by the main engines. In vessels of modern construction it may be assumed that:

	IHP
Air pumps absorb	0.001-0.002
Circulating pumps absorb	0.002-0.005
Feed " "	0.005-0.006
Bilge " "	0.001

When fans for the combustion are also driven by the main engines they absorb about 0.005 to 0.015 of the *IHP* estimated for the propulsion.

III. HORSE POWER ESTIMATED BY MEANS OF COEFFICIENTS OF PROPULSION.

Much of the practical value of experimental investigation is sometimes lost owing to the difficulty of assuming a correct value of the ratio $\frac{EHP.}{IHP.}$. This, no doubt, partly accounts for the favor the so-called *IHP* constants enjoy with most designers, especially with those more conservative in their practice, but it is not a sufficient reason to advocate the purely empirical methods of calculation of the *vieille école*. The use of constants, in experienced hands, may be of great help to designers, provided they possess much data upon which they can rely, because it is nothing but the comparison of a projected ship to others which have been constructed, and whose trial results are at hand.

The use of *IHP* implies the direct comparison of power instead of resistance, that is, of mechanical work instead of forces, based on the law of similitude, and the conditions under which this comparison is possible should not be lost sight of: geometrical similitude of forms, corresponding speeds of ships and propellers, and, as far as possible, engines of the same type.

What speaks in favor of these empirical methods is the fact that they are founded on a rational basis, and that, when comparing ships, it is not necessary to subdivide the total resistance into its elements.

Owing to the fact that all power curves present the characteristics of parabolas, and to the traditional assumption that the resistance of vessels varies with the second power of the speed, passing from the notion of resistance (force) to that of power (mechanical work), all formulæ giving the power necessary to propel a ship at a desired speed may be put under the general form:

$$IHP = K\ D\ V^3.$$

Constants deriving from this or any similar expression may have a practical value only when it is distinctly understood that such expressions apply only to speeds comprised in a very limited field. Instead of the displacement D alone, other geometrical elements of the vessel may be considered at the same time. The constants

$$G_1 = \frac{AV^3}{IHP},\ G = \frac{D^{\frac{2}{3}}V^3}{IHP},\ G_2 = \frac{SV^3}{IHP}$$

where A is the area of the midship section in square feet, S the wetted surface in square feet, D the displacement in tons, and V the speeds in knots, have been calculated for different ships of the Italian navy* and given in Table III. The table gives comparative values of these constants for a number of vessels of every description, more values of the well-known admiralty constant G being given for a greater number of ships in Tables IV*a* and IV*b*.

TABLE III.

RELATION BETWEEN POWER AND SPEED. RESULT OF TRIAL TRIPS OF SOME ITALIAN SHIPS.

Name and Description of Vessel.	Displacement tons	Speed knots	I H P's Constants $G_1 = \frac{A.\ V^3}{I\ H\ P}$	$G = \frac{D^{\frac{2}{3}}\ V^3}{IHP}$	$G_2 = \frac{S.\ V^3}{IHP}$
Battleship *Re Umberto*	13,500	19	764	230	14,000
"	13,500	15	839	250	15,500
"	13,500	10	732	220	13,700
Battleship *Morosini*	11,000	16	775	235	14.100
"	11,000	12	689	215	12,400
"	11,000	10	635	195	11,300
Battleship *Garibaldi*	7,400	19.5	646	205	14,600
Cruiser *Marco Polo*	5,000	15	623	260	17,600
"	5.000	12	818	295	20,000
"	5,000	9	993	325	22,000
Cruiser *Fieramosca*	4,000	13	603	210	14,000
Cruiser *Etruria*	2,500	19	495	170	11,500
"	2,500	17	603	200	14,200
"	2,500	10	807	270	18,300
Cruiser *Minerva*	850	19	431	170	14,500
"	850	15	538	215	18,300
"	850	10	570	230	20,000
Destroyer	250	30	419	206	18 000
"	250	25	377	180	15,800
Torpedo-boat	150	26	371	190	16.150
"	75	20	248	125	10.400
"	75	16	312	160	13.450
Transport *Trinacria*	8,500	16	656	248	17,450
Training ship *Vespucci*	2,700	13	678	205	21,300
Transport *Europa*	1,500	8	560	200	.....
Tug-boat *Atlante*	800	13	323	100	7.650
Tug-boats	120	9	290	85	4.950
	30	9	...	80	.....
Steam Launches	5	7.5	...	65	.....

Next in importance to the admiralty formula is the so-called French formula giving the constant $m = \sqrt[3]{\frac{A}{IHP}} \times V$.

It may be worth while mentioning a formula worked out by Mr. O. Richter, which, combining the advantages of the English admiralty formula and of the French, may be of aid in all those cases where speedy calculations are of more use than great

*The data are taken from Scribanti's *Lezioni di Teoria della Nave, II.*

accuracy. Mr. Richter considers a prismatic body in which the length, depth, and displacement remaining constant, the midship section varies: he finds that the resistance becomes a *minimum* when

$$A = \frac{Displacement}{0.6\ L}.$$

This he calls the most favorable midship section; variations of 10 per cent. of this section only produce variations of 1 per cent. in the resistance of the prismatic vessel. He therefore suggests introducing this value in the French formula, which may be therefore written:

$$IHP = \frac{V^3}{m^3} \times \frac{D}{0.6L},$$

or

$$IHP = \frac{V^3}{K} \times \frac{D}{L}.$$

Adopting this formula, where
IHP is the indicated horse power,
V the speed in knots,
D the displacement in tons,
L the length of the ship in feet,
K a constant,
it may be found very useful for rapid calculation, as the constant *K* is easily obtainable from ships similar to those projected.

The last three mentioned formulæ may, if used by experienced designers, furnish with some approximation good results, and may serve, when making estimates, to compare a projected ship to one of which reliable speed and power data are obtainable.

The following tables, IV*a* and IV*b*, contain the values of the constants *G*, *m* and *K*, calculated for a great number of vessels of every description, each represented in the tables by a Roman figure, and of which the actual speed trials have been considered. Most of the vessels chosen are of the latest design, and every care has been taken to get the most reliable information obtainable. For some ships it has been found interesting to give the results of progressive trials, in order to show how the constants vary for small variations of speed and power. Almost every class of vessel, naval and mercantile, is represented in these tables, and together with the relation between power and speed, the principal geometrical features of the ships are given as follows:

Displacement in tons.
Length at the water line in feet.
Area of midship section *A* in square feet.
And the coefficients of fineness:

$$\text{Block coefficient } \delta = \frac{\text{displacement volume}}{\text{length} \times \text{breadth} \times \text{draft}},$$

$$\text{Prismatic coefficient } \varphi = \frac{\text{displacement volume}}{A \times \text{length}},$$

$$\text{Midship section coefficient } \beta = \frac{A}{\text{breadth} \times \text{draft}},$$

$$\text{Water-line coefficient } \alpha = \frac{\text{w. l. area}}{\text{length} \times \text{breadth}}.$$

TABLE IV*a*.—RELATION BETWEEN POWER AND SPEED. TRIAL RESULTS OF MERCHANT SHIPS.

Description of Vessel.		Geometry of Vessel.							I. H. P.	Speed. Knots.	Third Power of Speed.	Approximate Values of:		
		Displacement. Tons.	Length at the W. L. Feet	Area of Mid-sect. Square Feet.	Coefficients of Fineness δ	φ	β	α				G.	m.	K.
1		2	3	4	5	6	7	8	9	10	11	12	12	13
Ocean liners.	I.	22,600	663	1,792	0 631	0.665	0.949	0.749	37,800	23.52	13,011	274	8.51	10.8
	II.	21,050	625	1.600	0.627	0.675	0.928	0 739	28,000	22.00	10,648	289	8.48	12.8
	III.	17.300	523	1,493	0.746	0.650	0.958	0 859	7,200	15.00	3,375	314	8.90	15.5
	IV.	11.350	502	1,232	0.579	0.641	0.902	0.732	16,100	19.80	7,762	244	8.42	11.0
	V.	8,960	436	1,092	0.598	0.650	0.920	0.769	8,800	17.70	5,545	272	8.80	13.0
Cargo and passenger steamers.	*I.	24,200	561	1,894	0.773	0.806	0.960	0.869	5,400	13.50	2,460	377	9.48	19.6
	II.	19.400	561	1,505	0.761	0.815	0 933	0.870	5,500	13.0	2,167	285	8.41	13.6
	III.	17.200	522	1,493	0.741	0.772	0.961	0.834	9,000	16.5	4,492	332	9.07	16.6
	IV.	20 800	500	1,752	0.782	0.829	0 943	0.876	4,000	12.0	1,728	327	9.14	17.8
	*V.	18,750	500	1,670	0.757	0.786	0 963	0.858	3,600	12.6	2,000	392	9.76	20.9
	VI.	17,900	500	1,543	0.776	0.803	0 958	0.860	5,000	13.0	2,197	300	8.77	15.7
	VII.	11 350	453	1,150	0.679	0 721	0.941	0.798	5,000	13.9	2,685	272	8.54	13.5
	VIII.	9 660	429	1,010	0.710	0.764	0 929	0.871	3,600	13.9	2,685	338	9.22	16.8
	IX.	12,190	429	1,241	0.767	0.800	0.958	0.855	2,500	11	1,331	281	8.74	15.1
	X.	11 910	420	1,220	0.780	0 822	0.949	0.843	2,700	11	1,331	257	8.45	14.0
	XI.	9.900	406	1,110	0.697	0.723	0.963	0.820	3,200	13	2,197	316	9.15	16.7
	XII.	9,170	381	1,048	0.754	0.795	0.948	0 857	2,800	12	1,728	270	8.65	14.8
	XIII.	7.760	344	966	0.798	0.8 6	0.966	0.887	1.700	11	1,331	307	9.10	18.2
	XIV.	3 345	262	645	0.655	0.699	0.937	0.817	700	9	729	232	8.75	13.3
	XV.	2,320	251	413	0.735	0.786	0.936	0.861	930	10	1,000	189	7.63	9.9
	XVI.	1,730	207	402	0.698	0 744	0.940	0.850	440	9	729	239	8.74	13.8
	XVII.	864	174	262	0.618	0.689	0.927	0 804	401	9.265	795	179	8.05	9.8
	XVIII.	926	151	316	0.720	0.730	0.980	0.858	350	8	512	139	7.76	9.0
Passsengers' excursion steamers.	I.	5 620	400	686	0.634	0.717	0.844	0.790	3,700	15	3,375	288	8.56	12.8
	II.	667	172	226	0.533	0.605	0.882	0.761	1,018	13	2,197	104	7.90	8.3
	III.	150	98	96	0.380	0.533	0.712	0.682	300	11.23	1,416	133	7.67	7.2
	IV.	129	106	74	0.578	0.593	0.975	0 717	225	12	1,728	195	8.30	9.3
	V.	65	85	44	0.447	0.607	0.738	0.687	250	12	1.728	111	6.73	5.3
Pilot boat.	I.	360	115	170	0.485	0 657	0.740	0.770	300	9.1	753.6	127	7.54	7.9
Tugs.	I.	334	110	.....	.....	.....	.....	.....	649.6	12.39	1,902	140	....	8.9
	II.	334	116	173	0.465	0.581	0.800	0 717	517	12.1	1,771	164	8.42	9.9
	*III.	46.8	49.2	56	0.485	0.606	0.800	0.760	150	9.1	753.6	65	6.55	4.8
	IV.	41 7	50.5	46.4	0.401	0.545	0.735	0.714	95	9	729	92	7.46	6.3
Steam launches.	I.	24.2	49.2	26.9	0.492	0.656	0.750	0.725	58	8	512	71	6.21	4.3
	II.	23.1	49.2	27.6	0.484	0 620	0.770	0 660	63	8.5	614	76	6.46	4.6
	III.	22.7	41.0	32.8	0.475	0.624	0.761	0.770	50	8	522	80	6.97	5.7
	IV.	19 75	45.3	25.4	0.403	0.615	0.664	0.742	53	8.2	551	73	6.42	4.3
	V.	12.32	38.4	19.4	0.42	0.584	0.720	0 703	94	9.2	778.7	41	5.44	2.7
	VI.	11.80	42.6	15.9	0.429	0 605	0.708		13.3	7	343	134	7.15	7'1
									15.9	7.5	422	138	7.20	7.3
									20.7	8	512	129	7.04	6.9
									29.2	8.5	614	109	6.66	5.8
									41.9	9	729	90	6.26	4.8
									59.4	9.5	857	75	5.86	4.0
									80	10	1,000	65	5.60	3.5
Paddle steamers.	I.	248	177	69.5	0.702	0.714	0.985	0 781	540	13.5	2.460	180	6.82	6.4
	II.	97 5	133	34.4	0.764	0.774	0.989	0.784	250	11.1	1.368	115	5.72	4.0

*It has not been possible to ascertain if the trial results of the ships marked * are quite reliable.

The *IHP* constants, contained in Tables IV*a* and IV*b* are:

$$G = \frac{D^{\frac{2}{3}} \times V^3}{IHP};\ m = V \times \sqrt[3]{\frac{A}{IHP}};\ K = \frac{D}{L}\frac{V^3}{IHP}.$$

Formulæ are given to determine at once the horse power without having to find first the ship's resistance; the following, cited by Johow, is by Thornycroft.*

$$\text{Log } k = 3.65450 \ldots\ldots k = 4513.0$$
$$\text{Log } f = 2.10170 \ldots\ldots f = 126.4$$
$$\text{Log } n = 2.57978 \ldots\ldots n = 380.0$$
$$\text{Log } G = 2.20041 \ldots\ldots G = 158.6$$

The formula is supposed to give very good results, but it is rather too complicated for rapid calculation; the expression

$$\int \sin^{2.5} \alpha \, ds$$

TABLE IV*b*. RELATION BETWEEN POWER AND SPEED. TRIAL RESULTS OF WAR SHIPS.

Displacement of Vessel.		Geometry of Vessel.							I. H. P.	Speed. Knots.	Third Power of Speed.	Approximate Values of:		
		Displacement. Tons.	Length at the W. L. Feet.	Area of Mid-sect. Square Feet.	Coefficients of Fineness. δ	φ	β	α				G.	m.	K.
1		2	3	4	5	6	7	8	9	10	11	12	13	14
	I.	14,800	400	.....	.....	.....	.....	.....	3,058	11.6	1,561	308	....	18.9
									11,670	16.97	4,887	253	....	15.5
									15.556	18.04	5,871	228	....	13.9
Battleships.	II.	12,600	413	1,661	0.600	0.675	0.913	0.755	16,000	18	5,832	197	8.47	11.1
	III.	11,500	411	1,481	0.577	0.664	0.869	0.768	15,000	19	6,859	232	8.83	12.8
	IV.	11,000	396	1,483	0.570	0.657	0.865	0.704	14,000	18	5,832	203	8.54	11.6
	I.	11,800	440	.....	.....	.....	.....	.....	4.780	14.457	3,020	327	....	17.0
									16.510	20.45	8,242	258	....	13.4
									21,302	21.365	10,142	246	....	12.7
	II.	9,650	440	.....	.....	.....	.....	.....	4 711	15.676	3,850	370	....	17.9
									11.326	20.49	8,603	238	....	11.7
									22,189	22.38	11,209	229	....	11.1
		6,658		906.8					1,150	10	1,000	308	9.25	13.5
		6,652		905.7					1,475	11	1.331	319	9.35	14.0
		6,645		905.1					1,920	12	1 728	318	9.34	13.9
		6,638		904.2					2,480	13	2,197	312	9.30	13.7
		6,631		903.4					3.190	14	2,744	303	9.20	13.3
	III.	6,624	429	902.5	0.485	0.590	0.822	0.728	4.080	15	3.375	292	9.12	12.7
		6,617		901.4					5,160	16	4.096	279	8.96	12 2
		6.610		901.4					6.520	17	4.913	265	8.81	11.6
		6,603		899.3					8.200	18	5.832	248	8.60	10.7
		6,596		899.2					10 580	19	6,859	228	8.36	10.1
		6,589		898.2					13.820	20	8,000	203	8.06	8 9
First-class cruisers.		5,635		901.4	0.437	0.550	0.854		13,030	21.59	10,063	232	8.75	10.2
		5,904		937.5	0.477	0.554	0.860		14.100	21.51	9,952	230	8.74	10.3
		6 217		980.6	0 483	0.558	0.866		12,700	21.02	9,287	246	8.96	11.3
		6,223		981.1	0.483	0.558	0.866		11,400	20.48	8,589	254	9.05	11.7
		6.135		969.3	0-482	0.557	0.864		7,150	17.75	5.592	261	9.11	11.9
		5,904		947.2	0.477	0.554	0.860		6,970	17.67	5,517	259	9.06	11.7
		6.173		974.1	0,482	0.558	0.865		7 720	17.59	5 442	237	9.05	10.8
		5,904		937.5	0.477	0.554	0.870		6,950	17.54	5,396	253	8.98	11.4
	IV.	6,173		974.1	0 482	0.558	0.865		4.500	15.2	3,512	262	9.13	11.9
		5,904	401	937.5	0.477	0.554	0.860		4.230	15.17	3,491	270	9.12	12.1
		5,173		974.1	0 482	0.558	0.865		2,618	12.79	2,092	268	9.20	12.3
		5,904		937.5	0.477	0.554	0.860		2,368	12.51	1,958	269	9.20	12.1
		6,167		973.1	0.482	0.558	0.865		2,370	11.09	1,599	226	8.70	10.3
		5,904		937.5	0.477	0.554	0.860		2,338	11.66	1,585	221	8.61	10.0
		6,173		974.1	0.482	0.558	0.865		8,900	18.67	6,508	246	8.95	11.2
		6,095		963.4	0.481	0.557	0.863		8,950	19.32	7,211	269	9.21	12.2
		6,132		969.0	0.482	0.557	0.864		3,940	14.13	2,821	240	8.86	10 9
		6.132		969.0	0.482	0.557	0.864		4,000	15.25	3,546	298	8.91	13.5
Second-class cruisers.	I.	5,791	354	.....	.....	.....	.....	.....	2,636	13.5	2,460	300	....	15.3
									8,972	19.24	7,122	256	....	13.0
									12,781	21.09	9,380	236	....	12.0
	II.	4,261	325	752.3	0.094	0.610	0.810	0.706	8.000	18	5,832	191	8.21	9.6
	III.	3,915	361	651.6	0.478	0.586	0.814		9.850	20.53	8,653	218	8 32	9.5
	IV.	2,602	342	452.9	0.452	0.577	0.783	0.671	6,000	10	6,859	216	8.30	8.7
Coast defence ships.	I.	3,440	247	759.2	0.563	0.638	0.877		4,750	14.73	3.196	154	8.04	9.1
	II.	3.425	247	756.0	0.566	0.638	0.888		4,500	14.85	3,274	165	8.20	10.1
Scouts.	I.	2,019	325	351.1	0.516	0.618	0.835	0.629	5,400	19	6,859	202	7.66	7.9
	II.	1,344	246	317.7	0.455	0.636	0.715	0.667	2,808	17	4,913	212	8.24	9.6
	III.	1,054	185	.....					1,435	13.33	2,368	170		9.4
Torpedo boats and torpedo boat destroyers.	I.	64.8	115	33.1	0.472	0.595	0.793	0.676	800	18.5	6 332	127	6.30	4.5
	II.	81	118	42.4	0.460	0.566	0.813	0.525	1,206	20.7	8,870	137	6.79	5.0
	III.	88.4	124	44.9	0.471	0.554	0.851	0.625	1,350	21.5	9,938	143	6.93	5.2
	IV.	157.1	151	65.3	0.447	0.555	0.806	0.613	2,400	33.0	12 167	147	6.94	5.3
	V.	262	179	92.2	0.460	0.567	0.811	0.613	2,200	22.0	10,648	197	7.66	7.1
	VI.	358.5	214	96.1	0.465	0.610	0.762	0.713	5 000	26.0	17,576	175	6.89	5.9
	VII.	532	225	.....	.....	.....	.....	.....	6,957	25.6	16,777	157	....	5.7
	VIII.	542	225	.....	.....	.....	.....	.....	7,633	25.87	17,314	151	....	5.5
	IX.	541	220	.....	.....	.....	.....	.....	6,967	25.64	16,856	161	....	5 9
	X.	532	225	.....	.....	.....	.....	.....	7,119	25.57	16,718	154	....	5.6
	XI.	541	220	.....	.....	.....	.....	.....	7,028	25.72	17,014	161	....	6.0
	XII.	197	165	.....	.....	.....	.....	.....	2,871	25.296	16,188	191	....	6.7
	XIII.	197	165	.....	.....	.....	.....	.....	3,030	25.33	16,252	180	....	6 4
	XIV.	197	165	.....	.....	.....	.....	.....	2.926	25.44	16,465	188	....	6.7

$$IHP = KV\left\{\int \Omega \frac{3n}{2n+L} V^{1.7} + G\frac{n^{\frac{2}{3}}+3}{L^{\frac{2}{3}}+3} V^{3.7} \alpha \Omega\right\}.$$

In this formula:

V is the speed in knots,

$$\alpha\, \Omega \int = \sin^{2\,5} ds.$$

where α is the angle which the superficial element ds makes with the direction of the ship's motion, and the coefficients k, f, n, and G are to be determined as follows:

*See also Transactions of the Institute of Naval Architects, vol. x.

may be easily determined;** the length, L, is given in feet. In the Russian and French practice Afonassief's formulæ are often used, and in most cases these lead to very accurate results. These formulæ*** are:

**A similar expression is to be found in Rauchfuss' formula, and a method for calculating it is given by Johow.

***Congres d' Architecte et de Constructeurs Navies, 1900. M. Afonassief: Formules Pratiques pour le Mouvement du Navire.

$$IHP = 1{,}000\left(\frac{BD^2}{L^2}\right)^{\frac{1}{3}}\left(\frac{V_0}{A_0}\right)^{3.0},$$

and

$$IHP = \frac{1{,}000}{1-\varphi_0}\left(\frac{BD^2}{L^2}\right)^{\frac{1}{3}}\left(\frac{V_0}{A_0}\right)^{3.0},$$

the first to be employed when the propellers have no slip, the second in propellers with a slip.

In these formulæ:

D is the displacement in tons,

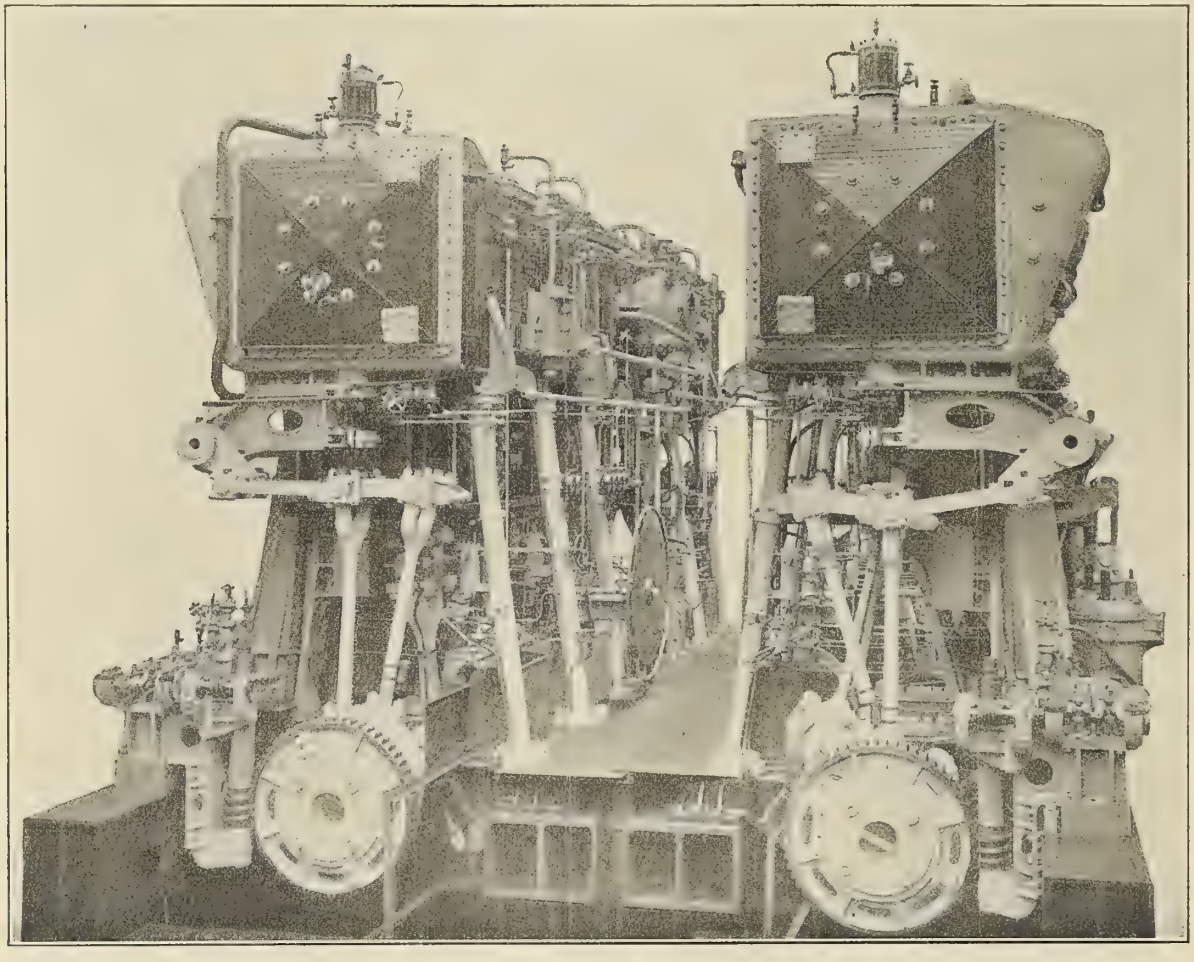
L the length at the W. L. in feet,

B the breadth molded in feet,

V_0 the speed in knots, and

A_0 a coefficient which may be assumed from *25.5* to *26*, for iron ships with bottoms clean and well painted, from *26.5* to *27* for iron ships with copper or zinc sheathing.

(To be continued.)



ENGINES OF THE BRITISH BATTLESHIP DOMINION, WHICH, ON TRIAL, DEVELOPED 18,438, AT 19.5 KNOTS SPEED.

Steam Trials of H. M. S. Dominion.

The first-class battleship *Dominion*, built for the British navy by Messrs. Vickers Sons and Maxim, Limited, at their naval construction works at Barrow-in-Furness, and supplied with her armor and armament from the same company's Sheffield works, completed recently the steam trials prescribed by contract. The results from every point of view were thoroughly successful. The speed attained at full power was, on a deep-sea course, 19.5 knots —one nautical mile per hour greater than was anticipated in the design; so that, excepting the *Triumph* and *Swiftsure*, she is the fastest battleship in the fleet, which, in view of her 16,400 tons displacement, and her powerful primary armament—four 12-inch and four 9.2-inch guns—is specially satisfactory. The power realized was 18,438 indicated horse power—438 indicated horse power over the contract requirement—notwithstanding that the machinery was worked under war conditions, with closed-in engine room and other restrictions, never before exacted on contract tests. The coal consumption at full power was 1.77 pounds per indicated horse power per hour for all purposes, while the water consumption for the main engines—17.3 pounds per horse-power hour—showed a satisfactory efficiency. If to these indications of success there be added the fact that the trials, themselves of 68 hours' duration, were carried out in four days, it will be conceded that the builders have once more proved their great ability.

The *Dominion* belongs to the *King Edward VII.* class, and has a length between perpendiculars of 425 feet, a beam of 77 feet 9 inches, and a depth molded of 42 feet 11 inches. At a draft of 26 feet 9 inches she displaces 16,400 tons. She has a belt, extending from the after armored bulkhead forward to the ram, 9 inches in thickness for the greater part of the length, reduced in three stages at the forward end to 2 inches. The maximum thickness extends along the citadel from 5 feet below the load line to about 3 feet above it; the next strake is 8 inches, and the top strake 7 inches, the latter reaching to the upper deck. The 12-inch guns are in barbettes of 12-inch armor, with hoods having sloping sides. The 9.2 guns are mounted separately in turrets of 7-inch armor on the upper deck, one at each quarter. The 6-inch guns are mounted within the 7-inch broadside armor in a concentrated casemate, as introduced first by the Vickers Company in the *Mikasa*. These 6-inch guns, five on each broadside, are on the main deck of 2-inch armor, while the upper deck is of 1-inch. On convenient positions for defense against torpedo-boat attack are fourteen 12-pounders, fourteen 3-pounders, and two Maxims. The armament includes also four submerged tubes for firing torpedoes.

Military tops have been dispensed with; but on the masts there have been constructed large observation stations which will carry Barr and Stroud range finders, and from these stations the guns will be trained and directed. The after navigating bridge has been abolished, although towers for search lights still remain.

There is, however, an admiral's bridge immediately abaft the main navigating station—a feature more usual on foreign war ships. As to ventilating cowls, the sails tried a year or two ago have been discarded, and the shafts to stoke holds, etc., are now fitted with mushroom-shaped covers, which can be raised and rotated on roller bearings. For the boat derrick on the main mast, vertical hydraulic engines are now substituted for the horizontal machines, not only because they require less deck room, but they are more efficient. The conning tower is larger; steam heating pipes and radiators are laid throughout the habitable quarters; baths are provided for all classes, and more attention has been paid to mechanical ventilation, especially in 'tween decks, as no port holes are possible in the armored sides of the vessel.

The engines are, as has been the case in battleships built since the *Duncan* class, ordered in 1898, of the cruiser type, with four cylinders arranged on the Yarrow-Schlick-Tweedy system. The high-pressure and intermediate cylinders, respectively 33 1-2 inches and 54 1-2 inches in diameter, have piston valves, and the two low-pressure cylinders at the fore-and-aft end of the sets are 63 inches in diameter, with double ported flat valves. These have balancing rings, and throughout the trials a vacuum of 20 inches was maintained behind them. The stroke of the engines is 48 inches, and the connecting rods are 8-foot centers. The ordinary double eccentric link motion is applied, the reversing gear being of the all-round type common to naval work, with a double-cylinder steam engine. All working parts are of forged steel, the crank shafts being 17 1-2 inches in diameter, with a 9-inch hole, the thrust and line shafts 17 inches, with a 9 1-2-inch hole, and the propeller shafts 20 1-2 inches with an 11 1-2-inch hole. The propellers, which are entirely of bronze, have four blades of 17 feet 6 inches diameter, with a pitch on trial of 18 feet 6 inches, and a developed surface of 86 square feet. All auxiliary machinery is independently driven, excepting the air pumps, which are two in number for each set of main engines, driven from the intermediate and high-pressure crossheads. There are four main condensers, with a total cooling surface of 19,000 square feet. The other auxiliary appliances include four evaporators, with an output collectively of 180 tons per 24 hours, and two distillers, having a total cooling surface of 107 square feet, to provide 45 tons of fresh water per 24 hours.

The boilers are of the Babcock and Wilcox type, sixteen in number, with 3 1-2-inch tubes, the total heating surface being 47,369 square feet, and the grate area 1402.5 square feet. There are four fire doors to each boiler, and as there are three boilers in a row in the aft and midship stoke holds, there are twelve furnace doors to be fired, the interval being three to five minutes. The firing was done only through one door in each boiler successively, so that the interval of renewing any portion of the fire in each boiler was five minutes. The cleaning of each section of the fire was at regular intervals of two hours, so that the entire grate in each boiler was cleaned every eight hours. Regularity in stoking and cleaning was ensured by clocks at each boiler, having an outer disc divided into sections to suit the intervals of firing—36 per hour. There is no steam-reducing valve. The feed pumps are located in each stoke hold and the water for the main engines was measured in a series of four tanks placed temporarily on the upper deck, independent tanks being provided for the auxiliary service. As shown in the table of results, the water consumption per indicated horse power per hour in the main engines was, on the 30 hours' trial at one-fifth power, 16 pounds; at 75 per cent. power, 16.6 pounds; and at full power, 17.3 pounds. When the auxiliary engines are included, there is apparently less economy at the lower powers; but this is due to the fact that then the auxiliaries are proportionately of much greater power to the total indicated horse power than when the engines are working at their highest speed. This same remark applies to the loss of water, which, on the higher power trials, was from 3 to 4 tons per 1,000 indicated horse power per 24 hours. This, however, had to be made up by the distillers, and the coal consumption is debited accordingly.—*London Engineering.*

STEAM TRIALS OF H.M.S. DOMINION.

Date of trial		October 28 and 29.			October 29, 30, 31.		November 1.	
Nature of trial		30 hours at 3,600 indicated horse-power.			30 hours at 12,000 indicated horse-power.		8 hours at 18,000 indicated horse-power.	
		Forward	Aft.		Forward	Aft.	Forward	Aft.
Draft of water		26 ft. 1 in.	26 ft. 8 in.		26 ft. 5¾ in.	26 ft. 11¼ in.	26 ft.	27 ft. 3 in.
Speed of ship		By log. 12.8 knots.			By log, 18.3 knots.		19.5 knots, measured course.	
Steam pressure in boilers		241.3 lb. per sq. in.			243.8 lb. per sq. in.		248.1 lb. per sq. in.	
Air pressure in stokeholds		Nil.						
		Starboard.	Port.		Starboard.	Port.	Starboard.	Port.
Vacuum in condensers		26.5	26.3		26.5	26.2	26.5	25.6
Revolutions per minute		78.3	77.5		113.9	113.7	125.8	125.5
Mean pressure in receivers	High	124.6	124.5		200.6	198.8	228 1	227.5
	Intermediate	26.6	25.5		64.5	61.6	84.3	83.1
	Low	1.6	1.9		11.0	11.4	21.7	21.0
Mean pressure in cylinders	High	35.5	38.0		79.5	80.1	90.7	91.5
	Intermediate	14.9	14.4		33.9	32.7	40.6	41.3
	Low forward	6.57	6.21		13 06	13 6	19.6	19.3
	Low aft	6.18	4.89		14.2	13.3	22.0	20.6
Mean indicated horse-power	High	561	630		1,934	1,945	2,434	2,453
	Intermediate	660	631		2,186	2,108	2,885	2,934
	Low forward	390	364		1,125	1,174	1,865	1 831
	Low aft	366	287		1,223	1,148	2,086	1,949
Total		1,977	1,912		6,468	6,375	9,270	9,168
Grand total		3,889			12,843		18,438	
Consumption of coal per indicated horse-power per hour		1.93 lb.			1.68 lb.		1.77 lb.	
Total loss of water per 1,000 indicated horse-power per 24 hours		5.34 tons			3.04 tons		3.38 tons	
Water consumption per I. H. P. per hour	Main engines	16.6 lb.			16 lb.		17.3 lb.	
	All purposes	21.6 "			18.4		18.3 "	

German Merchant Marine.—On April 1, 1904, the German empire possessed 1,011 ships, of an average capacity of over 1,000 tons each, against 976 ships of 1,000 tons in 1903. Of these ships, 786 were steamers and 225 sailing vessels. The increase in number occurred in steamers. To-day Germany has only 10 fast steamers; four, the *Fürst Bismarck, Augusta Victoria, Columbia,* and *Kaiserin Maria Teresa,* have been sold to Russia. Of the foregoing, 38 steamers carried the post; the others were passenger and freight steamers. In regard to size, the *Kaiser Wilhelm II.,* of the North German Lloyd, with its 20,000 tons, led. There are two steamers now in course of construction for the Hamburg-American line which will be even larger than the *Kaiser Wilhelm II.* One of these, the *America,* is being built in Ireland, and the other, the *Europa,* in the ship yard of Stettin. Of the number of ships in the Hamburg-American line there are 128 steamers. The North German Lloyd has 97 steamers; the Hansa, in Bremen, 42; the German Levant line, 30; the Wormann line, 29; the Hamburg-South American Steamship Company, 28; the Cosmos, 26, and the German-Australian Steamship Company, 25. There were 47 new steamers in course of construction in April, 1904. Of these, 13 were for the Hamburg-American line, 3 for the North German Lloyd, 6 for the Hansa, and 7 for the firm of H. C. Hahn and Schleswig. In the matter of sailing vessels, it is noticeable that the number is not increasing, but that there is a tendency to increase their tonnage capacity.—*United States Consular Report.*

COAST AND GEODETIC SURVEY STEAMER PATHFINDER.

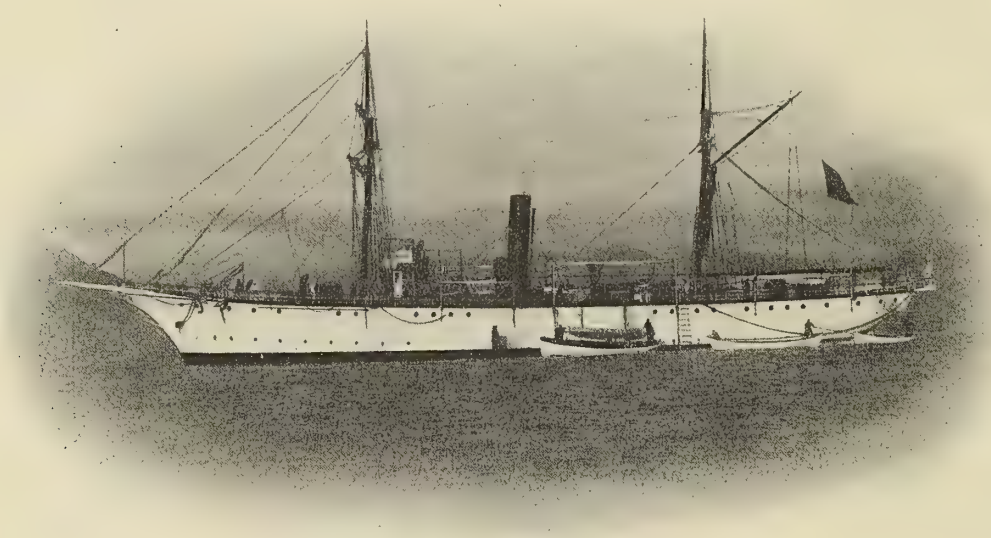
The *Pathfinder* was built at the Crescent Shipyard, Elizabethport, N. J., while Dr. Henry S. Pritchett was superintendent of the Coast and Geodetic Survey. She was launched on December 7, 1898, put in commission on April 23, 1899, and sailed for San Francisco, Cal., on June 7, arriving there September 18, 1899. The vessel is a steel, single-screw steamer, rigged as a brigantine with short bowsprit, and has a sail area of 4,478 square feet, sufficient to give her steerage way in a moderate breeze. She is fitted with bilge keels. The general dimensions are:

Length on water line at 10 feet draft	165 ft.
Length over all	193 ft.
Breadth of beam	33 ft. 6 ins.
Depth from top of flat keel plate to top of spar deck beams	19 ft. 8 ins.
Displacement at 10-foot water line	875 tons
Registered tonnage (gross)	690 tons
Registered tonnage (net)	469 tons

The spar deck stringers are of 15 and 10-pound plate, the main deck stringers of 10 and 8 1-2-pound plate, and the berth deck stringers and plating of 8 1-2 and 5-pound plate.

The spar deck beams, spaced 24 inches, and with a crown of 7 1-2 inches in 31 feet, are 3 by 5-inch angle bars secured to the frames by brackets of 10-pound plates. The main deck beams are straight; in the engine and boiler space they are 3 by 6-inch angle bulb, and forward and abaft this space 3 by 5-inch angle bars spaced 48 inches, except at mast partners, hatches, and bulkheads, where they are spaced 24 inches; they are secured to the frames by brackets of 8 1-2-pound plate. The berth deck beams are straight, spaced 48 inches, and of 3 by 4 1-2-inch angle bars, secured to the frames by brackets of 8 1-2-pound plate. The top of these beams is 7 feet below the top of the main deck beams at the center line of the vessel.

The spar and main decks are of 3 by 3-inch yellow pine, and the berth deck is laid with 2 1-2 by 4-inch yellow pine on top of 8-pound steel plates which are water tight. All hatch coamings are steel, and the berth deck hatches are plate steel and water tight when closed.



THE PATHFINDER, THE COAST AND GEODETIC SURVEY STEAMER, NOW IN THE PHILIPPINES.

The stem is of wrought iron 7 by 2 3-4 inches, and the stern post is of wrought iron 8 by 5 inches. The flat keel plate is of 20-pound steel and the vertical keel of 15-pound plate and 21 inches deep; the frames, spaced 24 inches, are Z bars 3 by 3 by 5 inches; the forward and after 12 frames being 3 by 5-inch angle bars. There are three longitudinals on each side—the first is continuous and water tight, under the engine and boiler space, and the other two on each side are intercostal of 10 and 12 1-2-pound plates.

The flat keelson plate, of 12 1-2 pounds, extends from frame 5 to frame 75; under the engine and boiler space it forms the top of the water tank, or double bottom, between the first intercostals; forward and aft this water tank the flat keelson plate is 9 inches in width on each side of the vertical keel.

The outside plating is of 15 pounds doubled at the water line forward and with an additional sheer strake of 15 pounds for a distance of 100 feet admiships; the horizontal seams are lapped and single riveted, the vertical seams are butted and double riveted. The lower courses of the thwartship bulkheads and coal bunkers are of 10 pounds and the upper course of 7 1-2 pounds steel plates stiffened by 2 by 3 inches and 2 1-2 by 3-inch angle bars of 4 and 6 pounds respectively, spaced 24 inches.

MACHINERY.

The machinery consists of a vertical, three-cylinder, triple-expansion engine 18, 27, and 44 inches and 28 inches stroke, driving a four-bladed screw 10 feet in diameter and 13 feet pitch. On the trial trip the indicated horse power was 847 with natural draft and 1,173 with forced draft. Under natural draft, with a steam pressure of 144 pounds and 115 revolutions, the speed was 12.4 knots, and under forced draft, with a steam pressure of 160.6 pounds and 121 revolutions, the speed was 13.4 knots. At a cruising speed of 10 knots the coal consumption is 11 tons in 24 hours.

The auxiliary machinery consists of a steam windlass and capstan, steam steerer, steam boat-hoisting winch, ice-making machine, two dynamos, bilge, fire, feed, and sanitary pumps, and an evaporator and distiller with a capacity of 1,600 gallons of fresh water per 24 hours.

There are two Scotch boilers each 11 feet long and 11 feet 1 inch in diameter, each with two corrugated furnaces; the combined grate surface is 90 square feet and heating surface 2,834 square feet. The boilers are tested for a working pressure of 165 pounds. One vertical donkey boiler 10 feet high and 5 feet in diameter with a grate surface of 15 1-2 square feet and heating surface of 574 square feet. The two main boilers are placed side

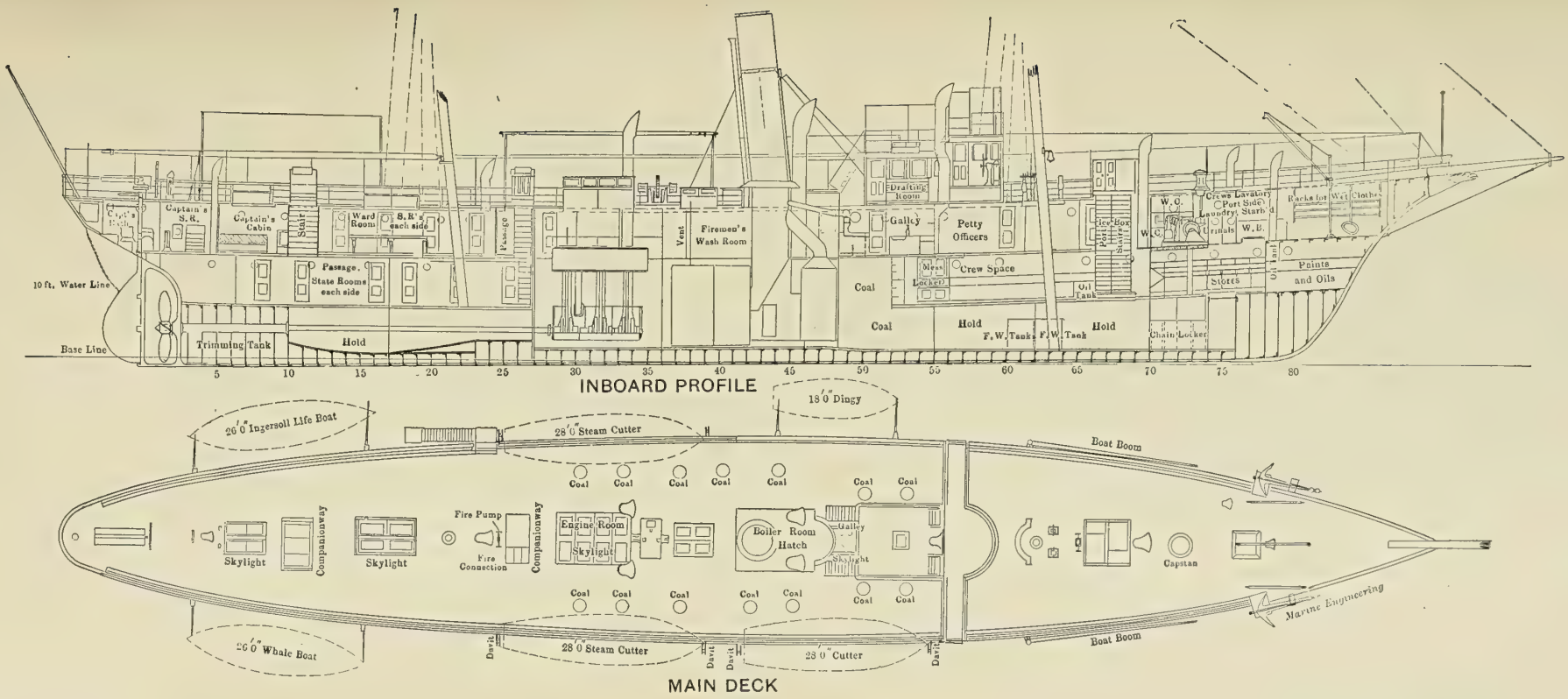


FIG. 2.—PLANS OF THE COAST AND GEODETIC SURVEY STEAMER PATHFINDER.

by side and fired from forward end. The capacity of the bunkers is 230 tons, which, at a speed of 10 knots, gives the vessel a steaming radius of 5,000 miles.

ARRANGEMENT.

The general arrangement is shown in Fig. 2 of the longitudinal and upper deck plans. Below the berth deck are two trimming tanks—one at each end of the vessel—and five thwartship water tight bulkheads, making five water tight compartments (holds). The five thwartship water tight bulkheads extend to the main deck and three of them to the spar deck. The fore-and-aft coal bunker bulkheads are also water tight.

The berth deck forward is taken up by a paint and oil locker abaft of which is the general (boatswain's) store room and brig. From the store room for a distance of 36 feet aft and the full width of the ship are crew berthing space and messing space for fifty men, allowing 100 cubic feet of space per man. On the berth deck aft are four state rooms, with six berths for junior officers; also eight state rooms and a sail room, with a passage leading fore-and-aft between them.

On the main deck forward of the water tight bulkhead for a distance of 30 feet to the bow are the lamp room, laundry, crew's lavatory, water closets, and the steam windlass; abaft this bulkhead for a distance of 42 feet to the fire room bulkhead (water tight) on the starboard side are the carpenters' tool room and four state rooms with a berthing capacity of ten petty officers; on the port side is the ice machine and refrigerator, office, sick bay with two berths, dispensary, petty officers' pantry, and photographer's dark room; at the after end, amidships, is the galley. The open space amidships on the main deck has hammock berthing for 26 men and is the mess room for the petty officers. On the main deck aft, just forward of the engine room bulkhead on the starboard side, are the dynamo room and engineers' tool and work room, and on the port side is the officers' water closet. Abaft the engine room bulkhead, abreast the ward room companionway on the starboard side, is the ward room pantry, and on the port side the ward room bath room. Abaft these is the ward room with eight state rooms, and abreast the cabin companionway is the captain's pantry, abaft of which is the cabin with two state rooms, toilet, and bath.

The engine, boilers, coal bunkers, dynamo room, fireman's wash room, and tool room occupy a space of 40 feet in length and the full height between decks to the spar deck, with 10 additional feet for coal bunkers under the berth deck.

SHIP'S OUTFIT AND STORES.

The ship's outfit of boats consists of two 28-foot steam launches, two 26-foot whaleboats, one 26-foot cutter, one 18-foot dinghy, and room for carrying a number of skiffs on deck. All the boats are rigged to be carried on the deck or the rail if required.

The vessel is lighted by electricity throughout and has a search light on the bridge; in the tropics electric fans are used for ventilation. There is a complete installation for heating the

vessel by steam, this being necessary for the health and comfort of the ship's company when working in Alaskan waters.

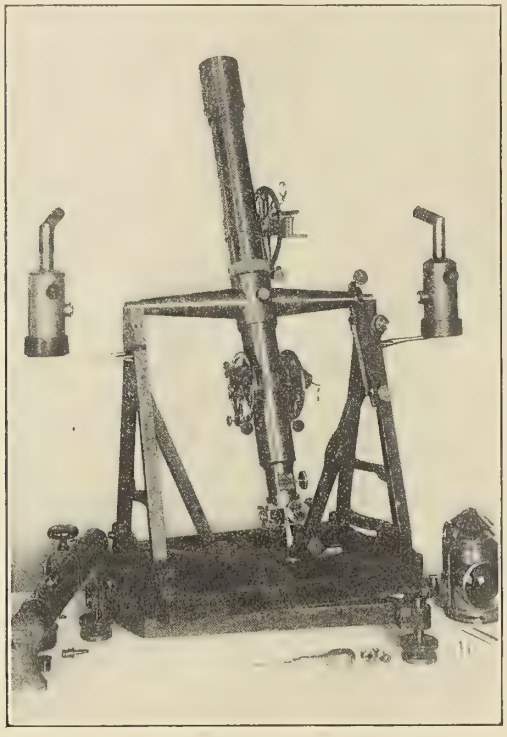


FIG. 3.—MERIDIAN TELESCOPE.

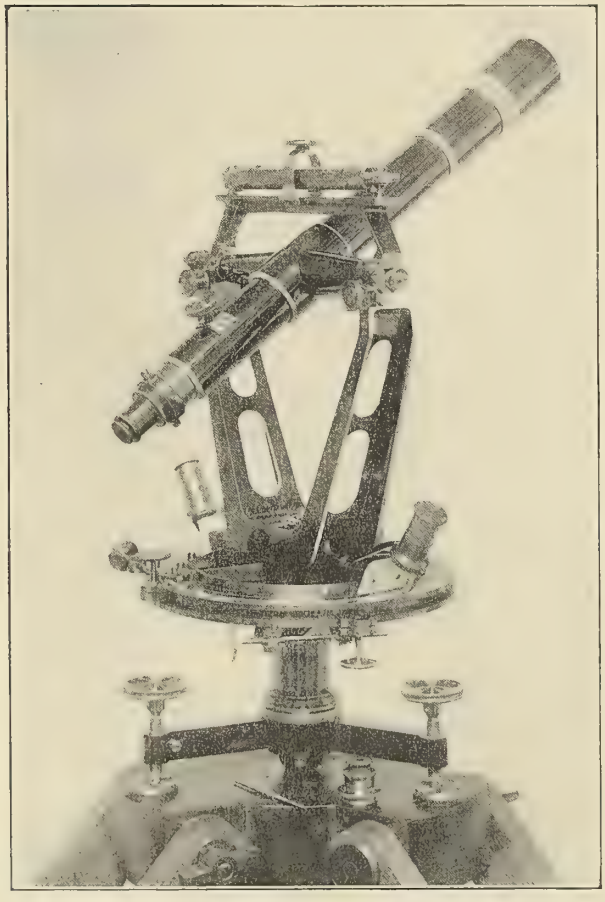


FIG. 4.—6-INCH GAMBEY THEODOLITE.

The water tanks have a capacity of 3,000 gallons, not including the tank under the engine room and boiler space, which has a capacity of 3,000 gallons. There are several spare propeller blades and spare parts of machinery which are most liable to be required in a long cruise. Ship's supplies, clothing, small stores, and provisions for 16 officers and a crew of 70 men for a six months' cruise can be stored in the holds and store rooms.

SURVEYING OUTFIT.

The outfit of surveying instruments includes eight mean time chronometers and two sidereal time chronometers, together with a meridian telescope for the astronomical determinations of azimuth latitude and longitude; three theodolites for triangulation; base measuring tapes with thermometers and spring scales;

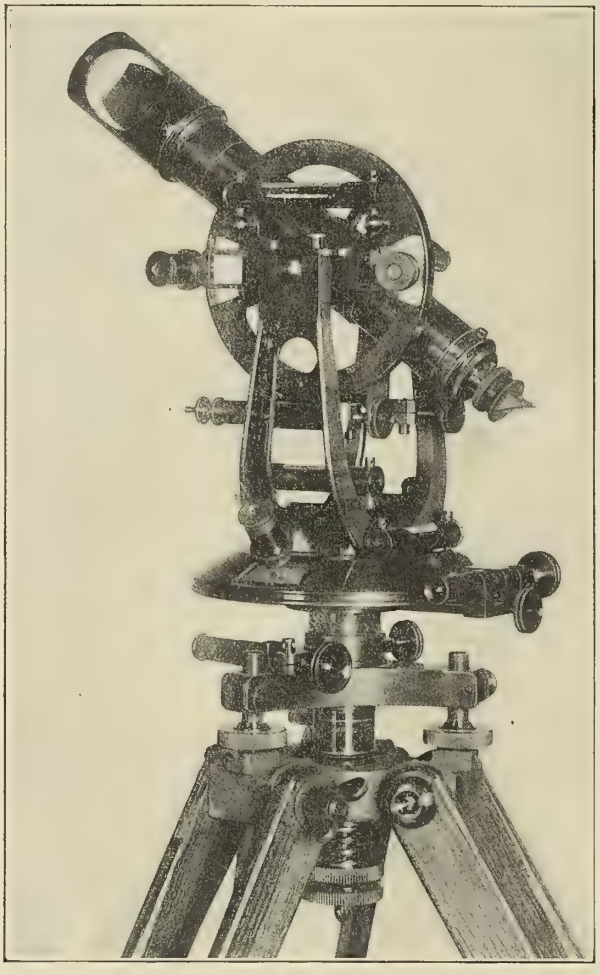


FIG. 5.—4-INCH THEODOLITE.

engineers' Y level; magnetometer; ordinary dip circle, and Creag dip circle with binnacle stand and compass for magnetic observations at sea; plane tables for topographical work; a complete outfit for topographic photography; sounding machines for deep sea and inshore hydrography; a channel sweep for finding small obstructions to a depth of 35 feet below the surface of the water; a marine sentry for use in unknown deep waters to give indications of rocks and shoals with a limited depth of water over them, and a complete outfit of instruments for the investigation of water temperatures and densities.

The sounding machines include a Sigsbee machine, a Lord Kelvin machine, three Tanner machines—two of the latter being fitted for use in boats; all of these have been in use for a number of years and have earned a high reputation for their satisfactory work. The Sigsbee machine, which is used for the greatest depths, is mounted on a platform over the stern of the vessel and far enough aft to prevent the sounding wire from fouling the screw. The machine has a record of sounding

in over 5,000 fathoms of water. The Lord Kelvin machine is mounted on the starboard quarter, and is used in depths of 100 fathoms or less while the vessel is under way and running at any speed. The Tanner machine can be mounted wherever found

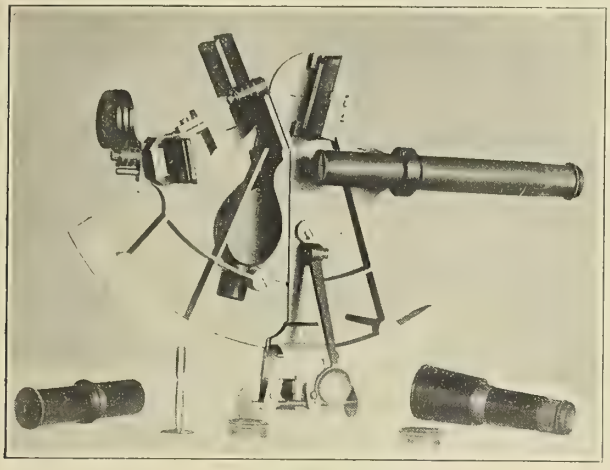


FIG. 6.—NAVIGATION SEXTANT.

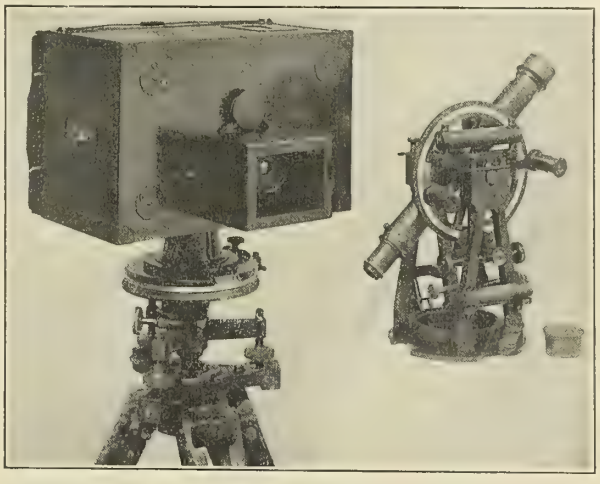


FIG. 8.—PHOTO-TOPOGRAPHIC CAMERA.

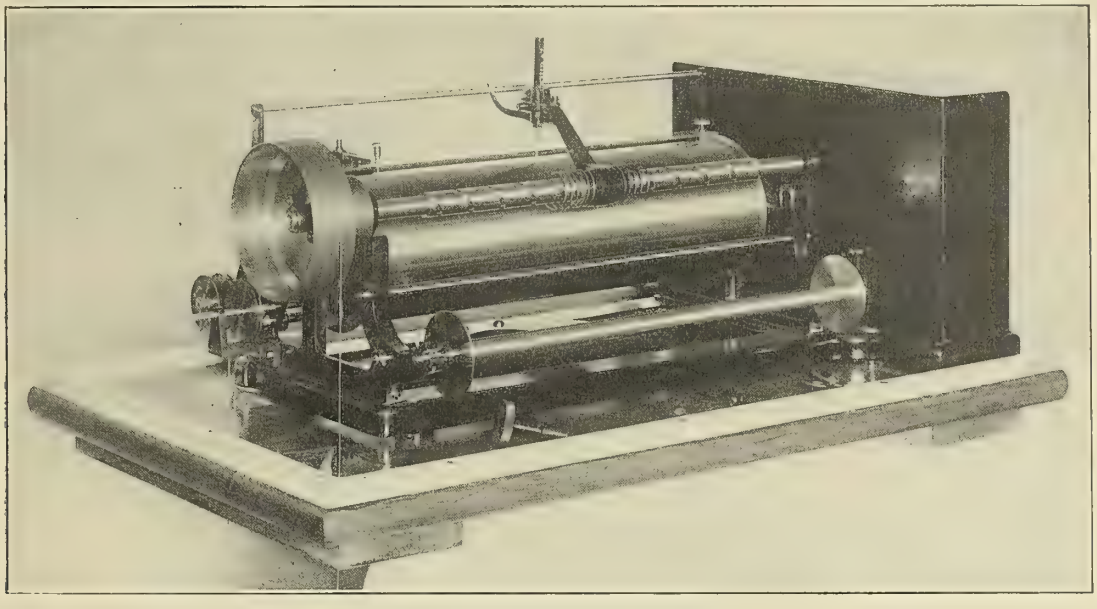


FIG. 9.—SELF-REGISTERING TIDE GAUGE.

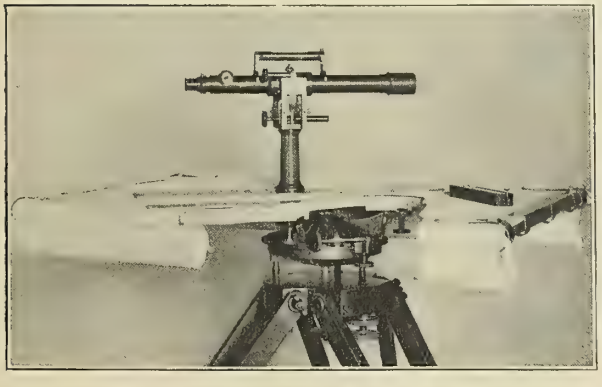


FIG. 7.—PLANE TABLE.

most convenient for sounding in depths of less than 100 fathoms. When sounding with the Sigsbee or Tanner machines the vessel's headway is stopped so as to get an up and down reading of the wire.

A more detailed description of the instruments is given herewith, with several illustrations, from which it will be seen the broad scope of work that the *Pathfinder* is designed for, as well as her extensive equipment.

Meridian Telescope. Fig. 3. For determining time (longitudinal), latitude, and azimuth. In beginning the survey of the Philippine islands a large number of geographic positions were determined astronomically at the shore terminals of the cable lines, many cables having been laid by the army incidental to the military operations. These positions served as initial points, or base stations, for extending the surveys in all directions.

Six-inch Gambey Repeating Circle. Fig. 4. Remodeled by the Coast and Geodetic Survey. A base line having been measured with a steel tape in the vicinity of one of the initial points, a triangulation is extended in the neighborhood, and in many cases reaches the shores of other islands. This instrument would be used in observing the angles, or in case of very long lines between the stations one of the larger ones in the cuts following.

Seven-inch Theodolite, for secondary triangulation and determination of time, latitude, and azimuth in difficult country and in rough mountainous country where the weight and bulk of the instrument is a consideration in transportation.

Ten-inch Repeating Theodolite, for triangulation and determination of azimuth.

Four-inch Theodolite. Fig. 5. For reconnoissance and tertiary triangulation.

Navigation Sextant, or Astronomical Sextant. Fig. 6. This well-known instrument is used in determining the position of the ship at sea by astronomical observations for purposes of navigation, and the more elaborate observations that may be taken in running deep-sea sounding lines.

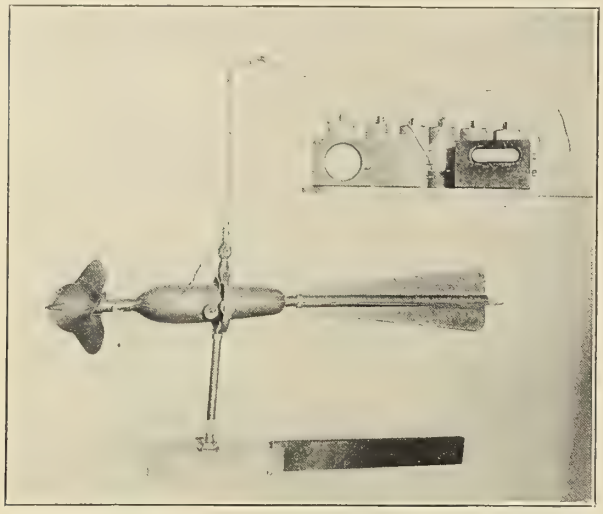


FIG. 10.—DEEP-SEA DIRECTION AND VELOCITY CURRENT METER.

The Plane Table. Fig. 7. Coast and Geodetic Survey form. After the triangulation has covered the ground, the shore line and topographical details, such as roads, houses, hills, etc., are developed with this instrument. In many cases where the survey may be desired immediately, a graphic triangulation is made with the plane table instead of waiting for the more precise work with the theodolite, the plane table stations being marked so that they may be recovered and included in the precise work when it shall be obtained.

Survey Photo-Topographic Camera. Fig. 8. Coast and Geodetic Survey type. This is a useful auxiliary to the plane table, theodolite, or transit, particularly in a mountainous country.

Self-Registering Tide Gauge, Stierle Form. Fig. 9. Improved by the Coast and Geodetic Survey. Soundings are taken at any stage of the tide and must be reduced to a common plane to be available for chart publication. The tide gauge records the water as it rises and falls, and if the observations are continued for a sufficient period they will furnish the data by which the time of the rise and fall can be predicted for several years ahead. Simultaneous observations on gauges at other places permit the predictions to be extended to those places, so that eventually from a few primary stations at which a long series has been observed, and numerous subsidary stations observed for short periods, tide tables can be constructed for all parts of the islands.

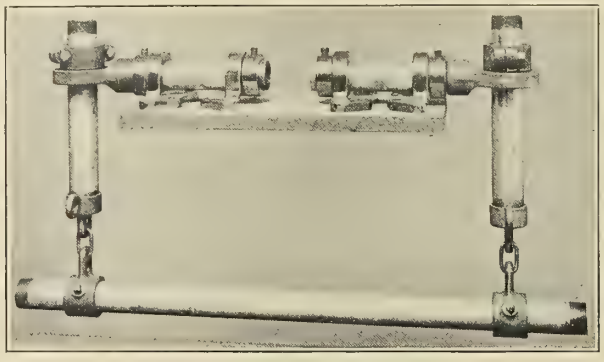


FIG. 12.—HARBOR AND CHANNEL SWEEP.

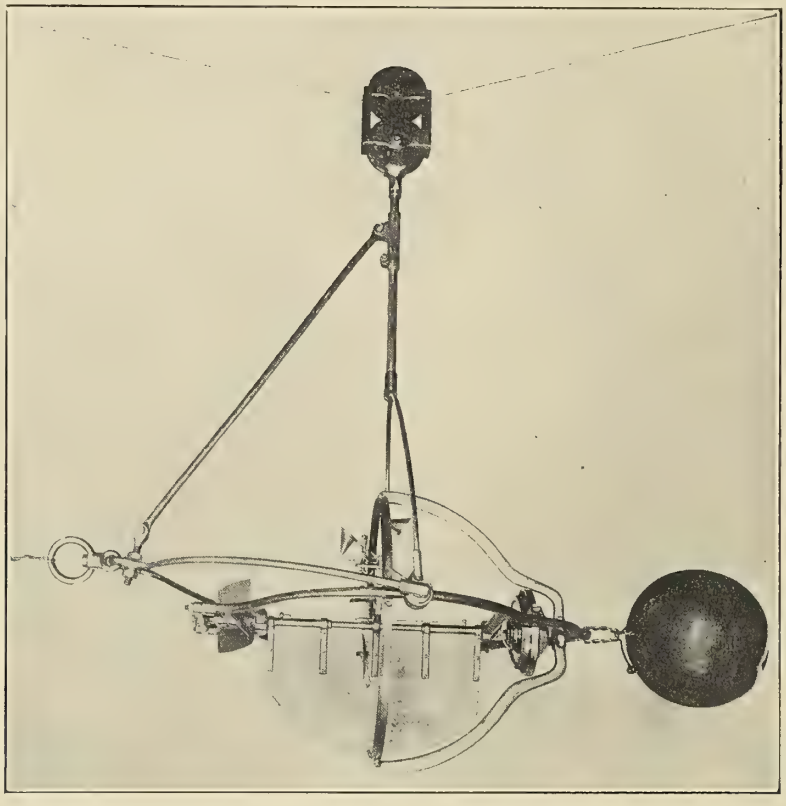


FIG. 11.—PILLSBURY DEEP-SEA METER.

The central drum is driven by a clock shown at the right in the illustration, while the shaft with pencil is revolved by floats by the raising or falling of the tide.

Ritchie and Haskell Deep-Sea Direction and Velocity Meter. Fig. 10. The meter holds a compass within its body that is connected electrically with a compass on the ship; the turns of the propeller are also recorded electrically on the ship. Therefore, when the meter is lowered in the water the direction or compass

course to which it points and the speed of the propeller are known on the vessel at all times. This meter is used to determine surface currents with only a slight difference in the arrangement of the tow lines.

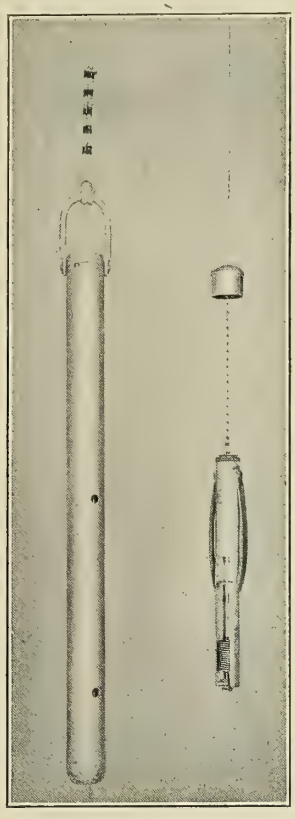


FIG. 13.—LORD KELVIN DEPTH RECORDER.

Pillsbury Deep-Sea Current Meter. Fig. 11. This meter was designed to measure the deeper currents of the ocean. It differs from the Ritchie and Haskell meter in principle. At whatever depth the observation is made the meter is locked before being drawn to the surface, and its markings read after being drawn on board. The currents of the Gulf stream in the Yucatan passage, off Key West, and across from Fowey rocks were measured with this meter.

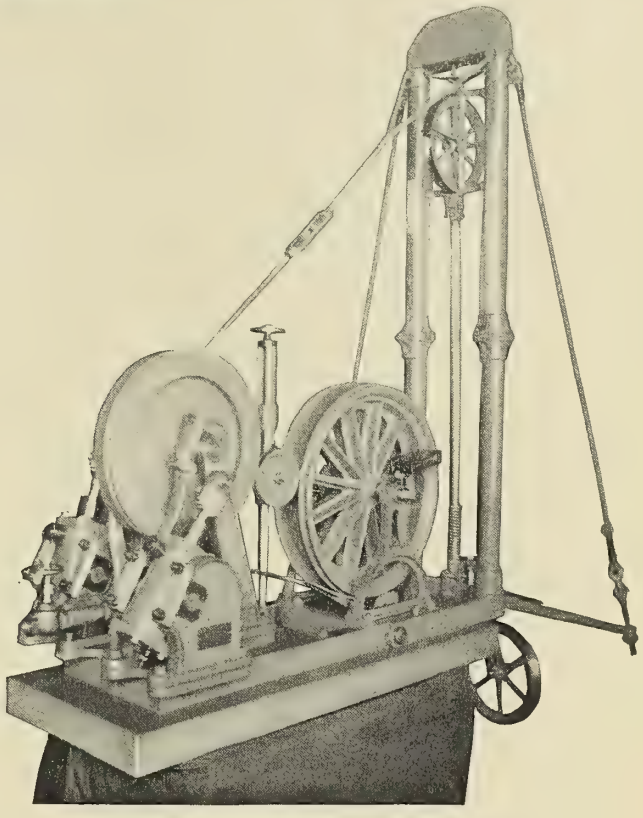


FIG. 14.—SIGSBEE DEEP-SEA SOUNDING MACHINE.

Bliss Taffrail Log. The propeller is dragged astern of the vessel and registers the speed of the ship in proportion to the turns of the propeller.

SOUNDING PARTY AT WORK IN A HARBOR.

Harbor and Channel Sweep. Fig. 12. The suspended bar is about 50 feet long and is lowered under the vessel to any desired depth not greater than about 35 feet. The two swivels fastened to the board are secured to the gunwale on the after part of the ship, and sufficient pipe is lowered through them to make the

desired depth. The piping is maintained perpendicularly by guys leading forward that can be readily cast off when the bar strikes an obstruction. The two links are made to break at a fixed strain to prevent tearing the apparatus to pieces. This apparatus increases the draft of the ship to any desired figure less than about 35 feet, and is used to determine how great a draft can be carried through a given channel without obstruction, or to find isolated rocks that have been reported or may be suspected to exist in a channel or harbor.

Sir William Thompson's (Lord Kelvin) Depth Recorder. Fig. 13. A device that determines the depth to about 80 or 90 fathoms without stopping the ship's headway to take a sounding. In this instrument the weight of the water compresses a spring; other devices similar in principle compress the air in a glass tube by which the depth is deduced. Their application is limited to less than 100 fathoms.

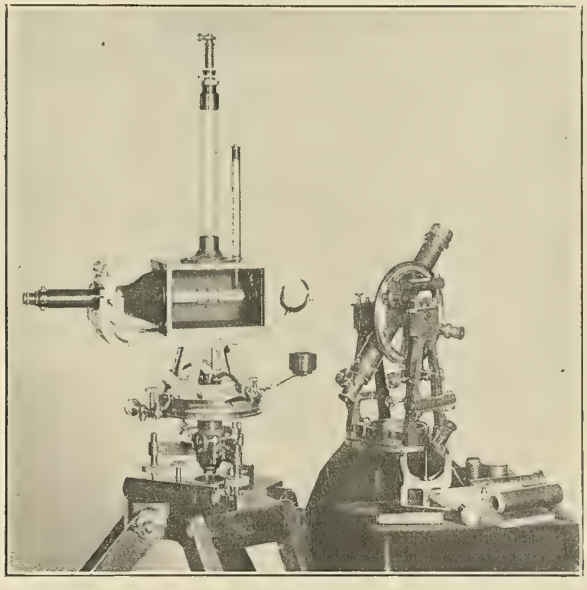


FIG. 15.—MAGNETOMETER. FOR DETERMINING MAGNETIC DECLINATION AND INTENSITY.

Sigsbee Deep-Sea Sounding Machine. Fig. 14. This machine has been used very successfully in securing soundings at very great depths as well as in lesser depths. The sounding line is piano wire to which a shot is attached by a simple device that detaches the shot when bottom is reached. A specimen cup is also attached to the sounding wire and slips through the shot when the latter is detached; bottom has been brought up in this cup from a depth of five miles. The machine has been used for developing the deeper depths of the oceans and for lines of "cable sounding."

The machine is constructed so as to lessen the strain on the sounding line from the rolling of the ship when the shot is at great depth.

Cosmos Boat Sounding Machine. This machine was devised on the Coast and Geodetic Survey launch *Cosmos,* and is well adapted to use on a launch for soundings of less than 500 fathoms. It consists of two geared wheels, one driven by a pair of handles and the other carrying a drum for winding the sounding wire. Its main advantages are the use of a light lead and quick recovery of the sounding wire; it reels in three fathoms with one turn of the crank handles.

Hydrographic Sextant. This instrument differs from the navigation or astronomical sextent in being much smaller and lighter, and that it reads only to half minutes.

Three-Arm Protractor. The position of the sounding boat is determined at frequent intervals by measuring two angles simultaneously with the sextant between three known positions on shore; these angles are then laid off on the protractor and plotted by the points on the sheet to which they were measured. The protractor shows the extension arms that may be added to make a longer reach.

Magnetometer. Fig. 15. For determining magnetic declination and intensity.

Dip Circle. Fig. 16. For determining dip of the needle, and, with the aid of compass mounted on top, also declination and intensity. The determination of the magnetic elements is a very essential feature of a survey for constructing a chart. A number of magentic observatories have been established in the United States and occupied for a series of years to determine these elements and their annual changes, and recently observations have been made by the ships on the ocean for the same purpose with such success that valuable results are expected in the future.

Since the *Pathfinder* has been in commission she has spent a winter season surveying in the Hawaiian islands, two summer seasons in Behring sea, and three years in the waters of the Philippine islands. In the latter waters she has made surveys, including all classes of work, of Romblon harbor, Romblon island, Ormoc harbor, Leyte island, Cebu harbor, Cebu island, San Bernardino strait and approaches from the sea, Albai gulf, east coast of Luzon, San Pedro bay and south coast of Samar island

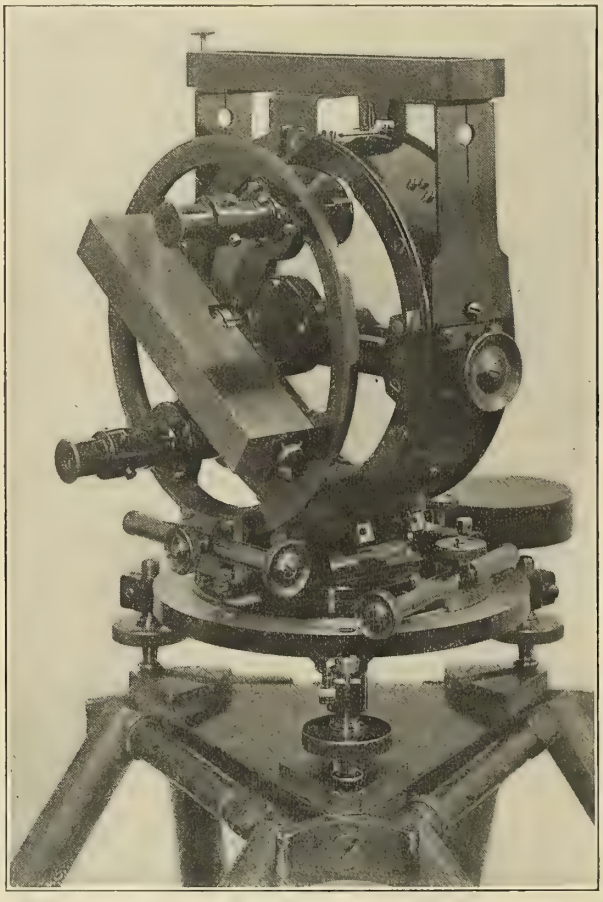


FIG. 16.—DIP CIRCLE FOR DETERMINING DIP OF THE NEEDLE, AND WITH THE AID OF COMPASS MOUNTED ON TOP, ALSO DECLINATION AND INTENSITY.

to Point Sangi, southwest coast of Leyte island from Illegos point to Maasin, Tobaco bay and Lagonoi gulf, east coast of Luzon island, supplementary work in Manila bay in the vicinity of Manila, hydrographic work along the west coast of Luzon island, north of Lingayen gulf, and has run a line of deep-sea soundings from a position off Point Bolinao to a point northeast of the Island of Formosa.

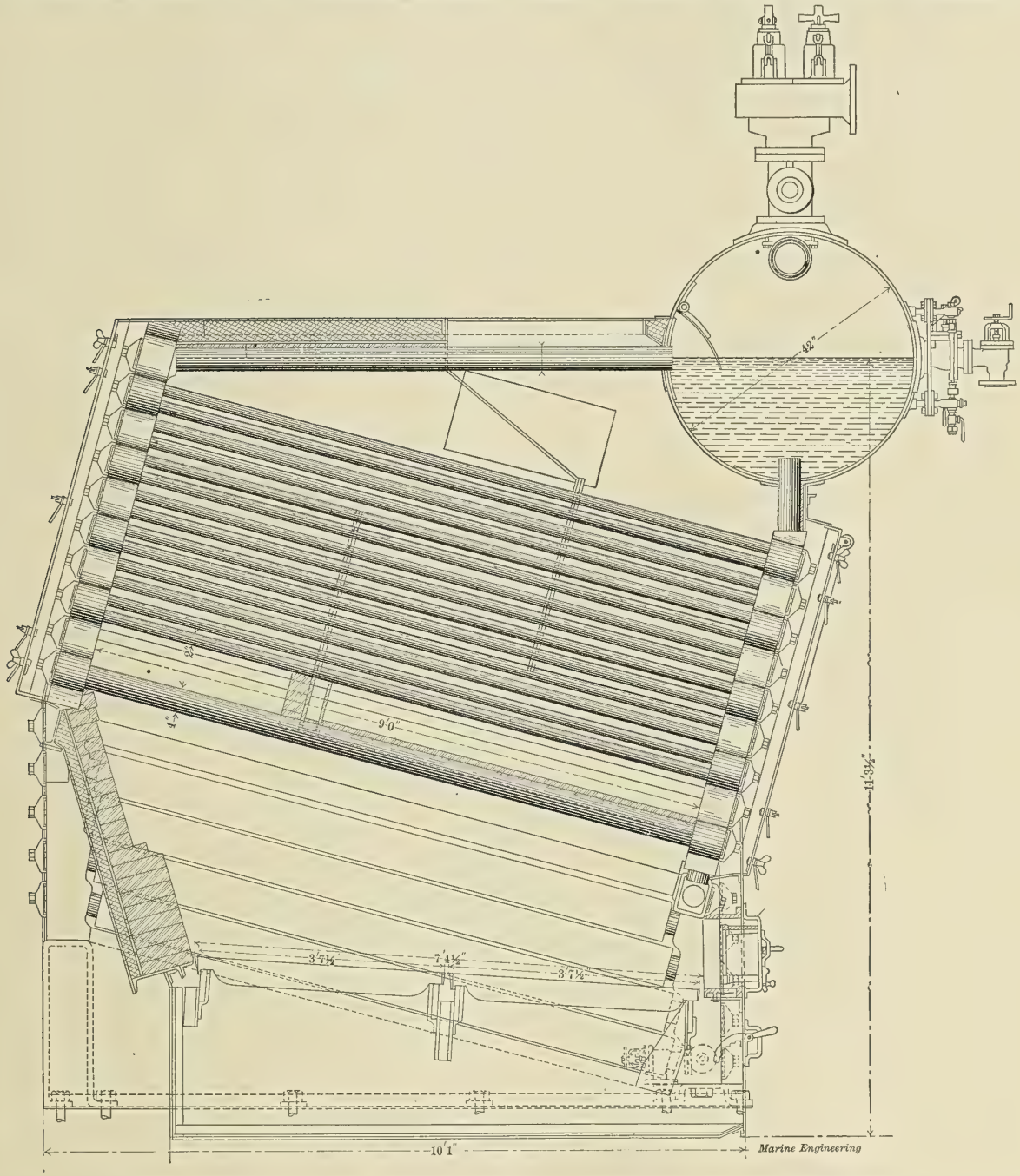
In all this work the *Pathfinder* has proved herself admirably adapted to the work for which she was designed.

FERRYBOATS FOR THE NEW YORK-STATEN ISLAND SERVICE.—III.*

MACHINERY.

The propelling machinery of these boats, now nearing completion, four of which are building by the Maryland Steel Company, and one by the Burlee Dry Dock Company, consists of two sets of inverted direct-acting, compound-surface condensing engines, with cylinders 22 1-2 and 50 inches diameter by 30 inches stroke. The two sets are coupled together with the low-pressure cylinders in the middle with cranks at an angle of 90 degrees. The high-pressures are at the ends with cranks opposite their respective low-pressure cranks. Each cylinder is cast with its steam chest, and each pair is firmly bolted together. The high-pressure cylinders are fitted with liners and one piston valve. The low-pressure cylinders each have two piston valves with separate liners. The cylinders are mounted on cast iron box-section back columns, and at the front are wrought steel columns.

The low-pressure pistons are of cast steel conical dished shape; the high-pressure pistons are of cast iron filled solid with lead. The piston rods are of forged steel, 5 1-2 inches diameter, and the connecting rods, which are 75 inches between centers, are forked at the upper ends, 5 1-2 inches diameter at the top, 6 1-4 inches at the bottom. The double forged steel crosshead pins are 6 1-2 inches diameter by 6 1-2 inches long. The bed plate of each engine is a single iron casting with a distance piece between engines. The crank shaft is 12 1-2 inches diameter, a solid forging for each engine. The crank pins are 12 1-2 by 12 1-2 inches, and the webs 9 1-4 inches thick. The line shaft is 12 inches and the thrust and propeller shafting are 12 and 12 1-2 inches respectively. Reversing is accomplished by a steam gear of the floating lever type, and the reverse shaft is 5 1-2 inches diameter. The valve gear is the Stevenson link type, with double bars 5 inches deep by 1 3-4 inches thick. The link blocks are of forged steel with pins 4 inches diameter by 3 1-4 inches long, and gibs of composition 12 inches long. The eccentric rods are of forged steel, and the straps of cast steel 3 3-4 inches wide lined with white metal. There is a thrust bearing at either end of engine of the usual horseshoe type, and cast iron stern tubes, one at



TYPE OF BABCOCK AND WILCOX WATER-TUBE BOILER TO BE INSTALLED ON THE STATEN ISLAND FERRY.

*Continued from page 474, October, 1904, issue.

either end of the boat, lined with lignum-vitæ staves. The propellers are to be of open hearth steel 11 feet diameter.

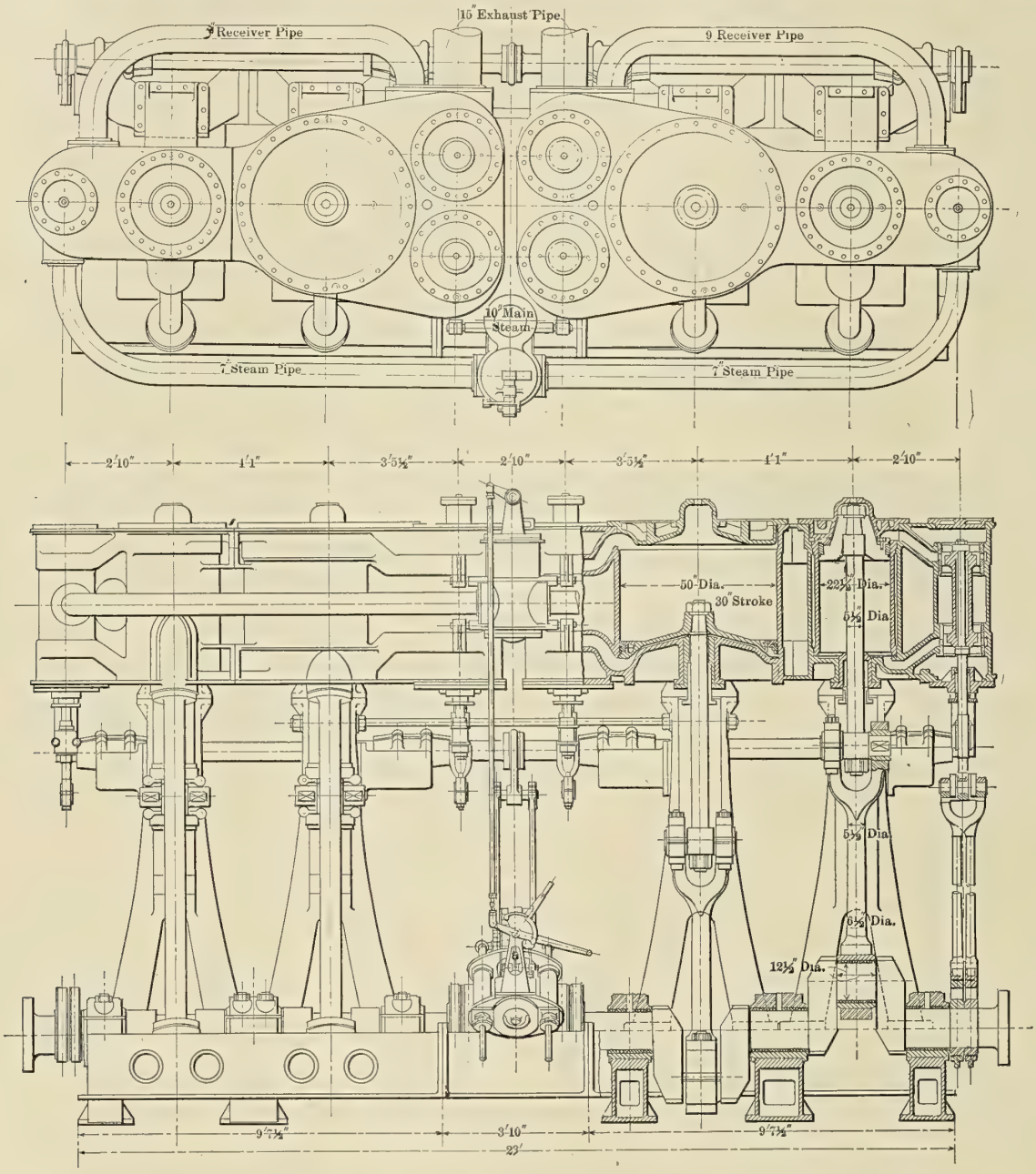
The exhaust from each low-pressure cylinder is led to back of the engine to one main condenser with sheet steel shell; 4,500 square feet of cooling surface, measured on the outside of tubes, is to be provided by 3-4 inch brass tinned tubes No. 18 B. W. G.

There will be one single-acting beam twin independent air pump, 10 and 22 inches by 18 inches. There is one centrifugal circulating pump with 12-inch suction driven by a 9 by 9-inch engine. There will be two main feed pumps of the vertical simplex type, 10 and 6 inches by 12 inches, and two auxiliary feeds 12 and 8 inches by 16 inches. The other pumps include duplex fire and donkey pump, 12 and 9 3-4 inches by 10 inches; a sanitary pump, vertical simplex, 6 and 8 by 8 inches; a fresh water duplex pump, 4 1-2 and 3 3-4 by 4 inches; injectors; bilge syphons, and feed-water heaters. A complete water service and lubricating system is to be fitted.

There are to be four boilers of the Babcock and Wilcox water-tube type in each ship, two placed in each of the fire rooms located at each end of the engine room, with an aggregate grate surface, with 7 feet bars, of 340 square feet, and an aggregate heating surface of 13,000 square feet.

The steam and water drums are 42 inches in diameter, the tubes being 2 inches in diameter, except the row immediately above the fire, which is composed of 4-inch tubes. The tubes are 9 feet long and of seamless cold drawn steel, all pressure parts are of forged steel, and the boilers will operate under 220 pounds pressure and are of the same type as those extensively used in recent United States war ships.

Forced draft is to be provided on the closed stoke hold system by two steel plate fans.



GENERAL ARRANGEMENT PLAN OF THE DOUBLE COMPOUND ENGINES FOR THE STATEN ISLAND FERRYBOATS.

Turbine Steamer Victoria.—The cable despatches reported last month that S. S. *Victoria,* which was equipped with Parsons steam turbines, had proved a great disappointment, so far as speed was concerned. The reporter must have been using his imagination, as we are informed by the Parsons Marine Steam Turbine Company, 97 Cedar street, New York, N. Y., that the turbines have not yet been fitted into the ship, so that there is no truth whatever in the report.

Scout Cruisers for the United States Navy.

Plans for the new scout cruisers, which have been under consideration in the Navy Department for several months, are nearing completion, and it is anticipated that they will be ready for distribution to prospective builders about the latter part of January. The Board of Construction has recommended that the three cruisers which were authorized at the last congress be of the following dimensions:

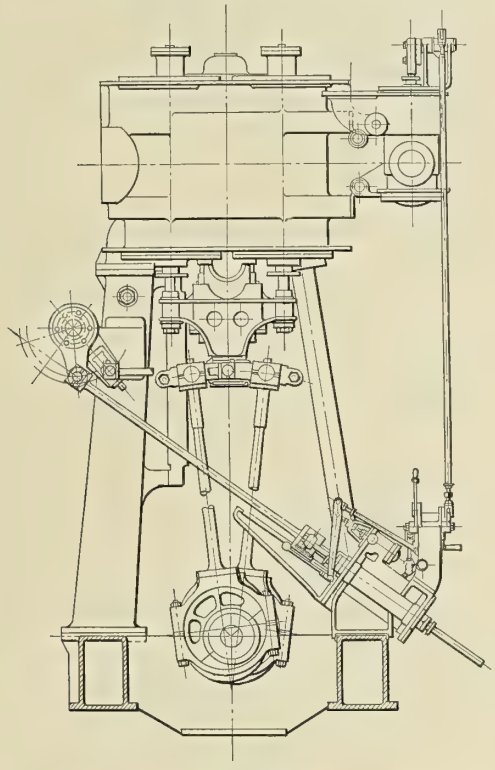
Length, between perpendiculars	420 ft.
Breadth	46 ft. 8 ins.
Draft, fully loaded	16 ft. 3 1-2 ins.
Deptn, amidships	36 ft. 5 15-16 ins.
Displacement, loaded	4,310 tons
Draft on trial	16 ft. 10 ins.
Corresponding displacement on trial	3,750 tons
Speed	24 knots

The battery is to consist of twelve 3-inch guns and two 21-inch submerged torpedo tubes, with 3,600 rounds of the 3-inch ammunition, and 8 torpedoes. The estimated weight of battery and full ammunition is 140 tons. Over the length of machinery space there is to be worked in an inclined nickel steel deck 1 1-2 inches thick, and a vertical steel protection deck 2 inches thick, extending above the tops of the boiler drums and cylinders of the main engines. At each end of the machinery space a thwartship nickel steel bulkhead, 1 inch thick, is to be fitted. Nickel steel armor, 2 inches thick, on the sides and 1 inch on the top, is to afford protection to that part of the steering gear which is above the water line. The total estimated weight of the nickel steel armor is 205 tons.

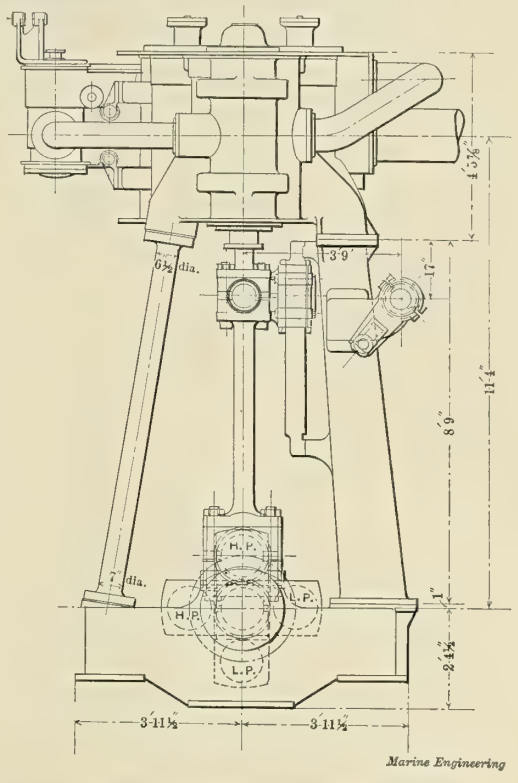
Four endless-chain electric ammunition hoists are to be installed, two forward and two aft, for supplying the ammunition for the 3-inch guns, while for handling the boats and coaling, etc., it is proposed to install four electric deck winches. The anchors are to be handled by one steam capstan windlass, and the steering gear is to be of the usual steam and hand type, with steering stations on the forward bridge and in the steering engine room. The vessel is to be lighted throughout by electricity, which power is also to drive the ventilation fans, from a large central generating plant of sufficient capacity to operate under battle conditions. The other electric equipment includes two large seachlights and wireless telegraphy outfit.

Two alternate proposals for powering the ships will be called for. One, which is according to plans prepared by the bureaus, is to include twin-screw reciprocating engines with necessary auxiliaries, with main engines designed to develop the maximum of about 16,000 indicated horse power on trial. Steam is to be generated by twelve water-tube boilers; besides the usual auxiliaries, there is to be an evaporating and distilling plant of 16,000 gallons capacity, a work shop, and refrigerating plant of two tons capacity. The total weight to be allowed for machinery, including spare parts, is 794 tons.

The alternate proposal will be for a vessel with the general characteristics of hull as set forth, but on the bidders' design of machinery, preference being given to a turbine installation, other things being equal.



Looking Forward at Low-Pressure Cylinder.



Looking Aft at High-Pressure Cylinder.

DOUBLE COMPOUND ENGINES FOR THE STATEN ISLAND FERRYBOATS.

Steam Turbines for the British Navy.

As a result of comparative trials of the four third-class, 3,000-ton cruisers recently completed for the British navy, very definite information is at hand concerning the efficiency of the marine turbine of the Parsons type at both cruising and full speeds. The turbine ship *Amethyst,* which underwent exhaustive trials in October and November of the past year, was built by Sir W. G. Armstrong, Whitworth & Company, Ltd., at the Elswick works, and supplied with steam turbines by the Parsons Marine Steam Turbine Company, Ltd. The other ships are the *Sapphire,* built by the Palmers Shipbuilding & Iron Company, Ltd., and the *Topaz* and *Diamond,* constructed by Laird, Cammell & Company.

The ships were designed for a speed of 21 3-4 knots, which, however, was exceeded in all cases, the maximum obtained by the reciprocating-engined vessels being 22.34 knots, while the *Amethyst* obtained a speed for four hours of 23.63 knots. By special arrangement of turbines for cruising purposes, the economy for the turbine at slow speed shows favorably in comparison with the reciprocating-engined vessels, while at the higher speeds a decreased coal consumption is in the neighborhood of 30 per cent. less than on the other vessels.

The British admiralty conducted extended trials on the *Topaz,* representing the three reciprocating-engined vessels, and on the

Amethyst. The dimensions of hull of the four vessels are identical, and will be found in Table I. The ships approach the scout cruiser class, although not of as high speed as intended for the latest vessels designed for this purpose. The armament includes four 4-inch guns with 4-inch armor hoods, and eight 3-pounders, three Maxim guns, and two torpedo tubes placed on the upper deck.

TABLE I.

	Amethyst.	Topaz and Diamond.
Length	360 ft.	360 ft.
Beam	40 "	40 "
Draft trial about	14 " 6 ins.	14 ft. 6 ins.
Displacement trial	About 3000 tons.	About 3000 tons.
Machinery	Parsons turbines 3 shafts	Twin screw 24 1-4, 38 1-2, 42 1-4, 42 1-4 24 ins.
Machinery total weight	535 tons.	537 tons.
Boilers	10 Yarrow.	10 Laird-Normand.
Grate surface	4.935 sq. ft.	
Heating surface	25,968 " "	26,000 sq. ft.
H. S. ÷ G. S.	52.5.	
Evaporation per sq. ft. H. S. per hour	7 07.	7.87
Propellers	3 Three bladed.	2 Four bladed.
Diam	Three 6 ft. 8 in.	
Pitch	Center 6 ft. 6.7 ins.	
"	Side 5 ft. 9 ins.	
Area each	Center 19.64 sq. ft.	
" "	Side 19.48 sq. ft.	

The unique arrangement of the turbines on the *Amethyst* permits driving them under most favorable conditions at different speeds. On the forward end of the port and starboard side shafts are placed a high- and intermediate-pressure *cruising turbine* respectively, each with drum 44 inches in diameter, but of different lengths and different arrangement of blades. Directly behind each of these turbines is, first, a low-pressure turbine 60 inches diameter, and at the rear end a backing turbine. The main high-pressure turbine with drum 60 inches diameter is placed on the center shaft. The piping is so arranged that the steam from the boiler can be led directly into either high-pressure turbine or into the intermediate turbine. Thus, at the lowest cruising speeds, steam is admitted to the high-pressure cruising turbine, thence it flows to the intermediate cruising turbine; thence to the main high-pressure turbine; finally to the two low-pressure turbines. At intermediate cruising speed the high-pressure cruising turbine is cut out, the steam entering the intermediate cruising machine, while for maximum speed both the cruising turbines are cut out, and the steam enters direct to the main high-pressure turbine on the center shaft. A forced lubrication system is fitted to the main journals.

Now, referring to the table giving the economic results of the series of trials of the two vessels, most surprising results are found, which, if they had not been semi-officially published in the *London Times* and *Engineering* might be questioned. The auxiliary machinery on the *Amethyst* exhausted direct to the condenser, and thus the low-pressure turbines did not receive the extra steam that the low-pressure cylinders of the reciprocating engines got from their auxiliaries. The effect of this on the general results will be appreciated by noting that on the *Topaz*, when running at 10 knots, 21 per cent. of the total steam was used by the auxiliaries alone, and when running at 14 knots the consumption was 13 per cent. or, 5,672 pounds of the total given in the table. Another point to be noted, is the easier steaming of the boilers on the *Amethyst*. The highest average pressure in the stoke hold for the two full-power trials was 1.65 inches; while that on the *Topaz* averaged 1.9 inches at full power. The results given in the right-hand column of Table II. are the averages of two 4-hour trials for each ship. It will be noticed here that practically 1.9 knots greater speed was obtained with the *Amethyst* on about 10 per cent. less coal. The horse powers for the turbine vessel are assumed to be the same as those of the *Topaz* for corresponding speeds, and while the total weights of machinery on the two ships are practically the same, this increase of speed, corresponding to an increase of almost 40 per cent. in estimated power, is attendant with the same weights as upon the *Topaz*. These results go to prove that the marine turbine when running at full speed is capable of an economy far in excess of other types of engines.

TABLE II.

WATER AND COAL CONSUMPTION TRIALS.

	Amethyst	Topaz	Amethyst	Topaz.	Amethyst	Topaz.	Amethyst	Topaz.	Amethyst	Topaz.
	24-hour trial at 10 knots		24-hour Trial at 14 Knots.		30-hour Trial at 18 Knots.		8-hour Trial at 20 Knots.		4-hour Trial at Full Power	
I. H. P.	897	897	2250	2251	4770	4776	7280	6689	13,500	9721
Speed, knots.	10.	10.058	14.062	14.08	18.186	18.069	20.6	20.063	23.35	21.465
Total steam per hour lbs	26,260	21,294	44,090	42,260	76,493	90,500	100,606	134,248	183,685	204,545
Steam per I. H.P. hour lbs.	29.3	23.74	19.6	18.77	16.	18.95	13.8	20.07	13.6	20.21
Total coal per hour lbs	2893.	2296.	4725.	4640.	8372.	10,900	10,937	15,451	24,223	26,915
Coal per I. H. P. hour lbs.	3.22	2.56	2.1	2.06	1.75	2.28	1.5	2.31	1.8	2.77
Evaporation per lb. coal lbs	9.1	9.3	9.35	9.13	9.15	8.3	9.7	8.7	7.6	7.75
Knots run per ton coal	7.42	9.75	6.6	6.8	4.8	3.7	4.22	2.9	2.16	1.83

Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, report under date of December 10, 1904, the following degree of completion of vessels building for the United States Navy:

BATTLESHIPS.			DEGREE OF COMPLETION. PER CENT.	
			Nov. 1	Dec. 1.
Ohio	18 knots.	Union Iron Works	100	
Virginia	19 "	Newport News Co	71.21	73.43
Nebraska	19 "	Moran Brothers Co	61.4	63.64
Georgia	19 "	Bath Iron Works	67.47	69.67
New Jersey	19 "	Fore River Shipbuilding Co	70.7	72.2
Rhode Island	19 "	Fore River Shipbuilding Co	73.5	75.1
Connecticut	18 "	Navy Yard, New York, N. Y.	56.4	58.71
Louisiana	18 "	Newport News Co	61.5	64.73
Vermont	18 "	Fore River Shipbuilding Co	25.8	29.5
Kansas	18 "	New York Shipbuilding Co	31.2	35.6
Minnesota	18 "	Newport News Co	46.56	50.24
Mississippi	17 "	Wm. Cramp and Sons	11.89	15.26
Idaho	17 "	Wm. Cramp and Sons	10.61	13.4
ARMORED CRUISERS.				
Pennsylvania	22 knots.	William Cramp and Sons	94.79	97.03
West Virginia	22 "	Newport News Co.	95.5	97
California	22 "	Union Iron Works	65	66.7
Colorado	22 "	William Cramp and Sons	97.11	98.42
Maryland	22 "	Newport News Co	92.16	93.41
South Dakota	22 "	Union Iron Works	63	64.1
Tennessee	22 "	William Cramp and Sons	54.58	58.71
Washington	22 "	New York Shipbuilding Co	50.2	55.3
PROTECTED CRUISERS.				
Chattanooga	16½ kts.	Lewis Nixon	97	99.13
Galveston	16½ "	Wm. R. Trigg Co	94	95
St. Louis	22 "	Neafie and Levy Co	54	56.6
Milwaukee	22 "	Union Iron Works	60	63.2
Charleston	22 "	Newport News Co	84.34	86.06
GUNBOATS.				
Dubuque	12 knots.	Gas Engine and Power Co	68.2	72.3
Paducah	12 "	Gas Engine and Power Co	64.9	68.4
TRAINING SHIPS.				
Cumberland	Sails	Navy Yard, Boston	80	85
Intrepid	"	Navy Yard, Mare Island	63	67.5
TRAINING BRIG.				
Boxer	Sails	Navy Yard, Portsmouth	90	95
TORPEDO BOATS.				
Stringham	30 knots.	Harlan and Hollingsworth	99	99
Goldsborough	30 "	Wolff and Zwicker	99	99
Blakely	26 "	George Lawley and Son	99	99
Nicholson	26 "	Lewis Nixon	99	99
O'Brien	26 "	Lewis Nixon	98	99½

MEXICAN GUNBOAT BRAVO, LAUNCHED NEAR GENOA, ITALY.

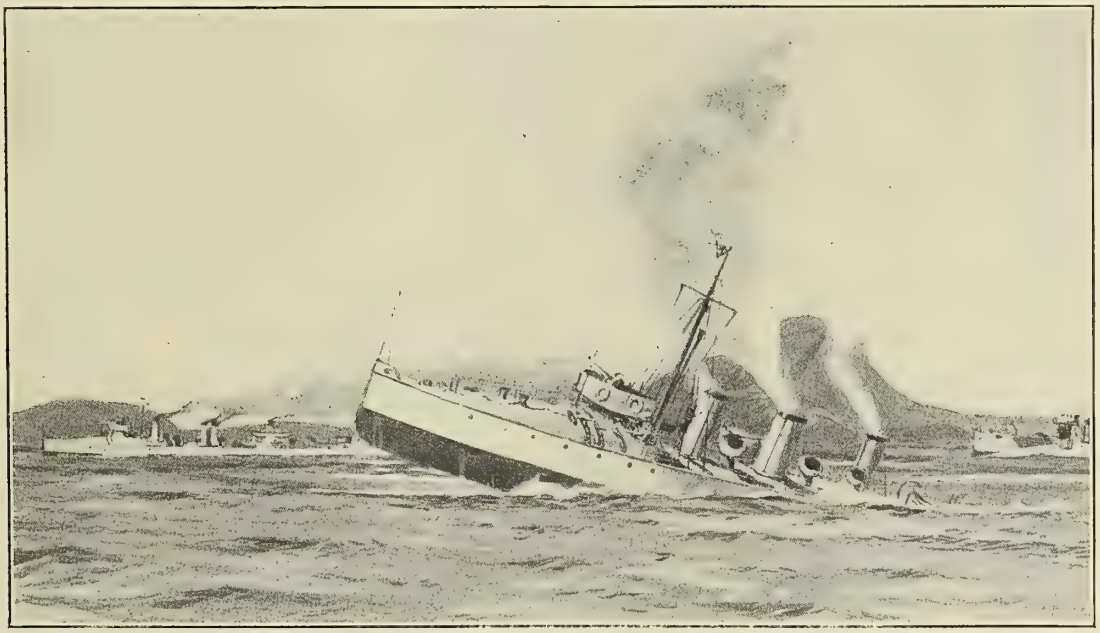
Mexican Gunboat Bravo.

There were recently delivered to the Mexican government two twin-screw gunboats, the *Bravo* and *Moreoa,* built at the Oderos ship yard at Sestri Ponente, near Genoa. The general dimensions are as follows: Length, 250 feet; beam, 34 feet; depth, 14 feet; displacement, 1,280 tons; designed speed, 17 knots. The vessels are very light draft, for navigating the shallow rivers of the Mexican coast, while at the same time the bunker capacity is sufficient to carry the vessel 5,000 miles at a speed of 10 knots. The propelling machinery consists of two sets of three-cylinder, direct-acting, triple-expansion engines of 2,600 I. H. P. Steam is supplied by two water-tube boilers.

There are two 4-inch R. F. guns of Bethlehem make, which fire directly ahead and astern respectively, and have a total arc of fire of 150 degrees; four 2 1-4-inch R. F. G., Schacidercant type, are fitted on broadside and one on the bridge. The 4-inch guns are supplied with ammunition by electrical hoists capable of delivering 24 charges per minute. Electrical hoists are provided for the other guns also. The crew will be composed of 26 officers and 90 men, and in addition quarters are provided on board for about 270 troops. There are six boats, two of which are steam launches, for quickly disembarking the troops. Fresh air is supplied to the quarters by electric fans, and a large refrigerating plant is installed. The *Bravo* will proceed direct to Mexico as soon as she has completed her trial trip.

British Destroyer Chamois Sinks.

While running at full speed, in company with nine other destroyers, off the coast of Greece, the *Chamois* met with a mishap to her machinery and sank in deep water. All of the crew escaped, but the exact cause of the accident could only be surmised. It is believed that the port propeller brackets gave way, carrying the shaft and propeller with them. The port engine immediately raced and wrecked itself before steam could be shut off. The after end of the hull was badly damaged, probably by the twisted shaft, or the propeller striking the thin hull plates, and the after compartments were immediately flooded. The vessel settled by the stern, and, as the thwartship bulkheads gave way under the water pressure, the machinery space was flooded, and the vessel sank stern first, as shown in the illustration, which is reproduced from London *Illustrated News.*



BRITISH DESTROYER CHAMOIS FOUNDERING IN THE MEDITERRANEAN AS A RESULT OF MACHINERY ACCIDENT.

Marine Engineering

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Notice to Advertisers.

Changes to be made in copy, or in orders for advertisements, must be in our hands not later than the 15th of the month, to insure the carrying out of such instructions in the issue of the month following.

IN taking a retrospect of shipbuilding for the year just closed, the only encouraging feature seems to be that it is apparently impossible for shipbuilding to go through a more depressed season than that of 1904. Three of the large over-sea freight carriers that have been under course of construction for a long time were finished, and there remains but one deep-sea steamer building at any American shipyard. The worst feature is that there are few new orders and but very little prospective work in sight. The policy of retrenchment observed by the shipyards for the last three years has about reached the limit, and it is to be feared that the depressing conditions may force the few remaining solvent companies into bankruptcy. The year has recorded the re-organization of the shipbuilding trust into the Bethlehem Steel Corporation; some of the smaller yards have applied for receivers, while a few others, which had been forced to close, have been able to re-open their gates. If it were not for the government work in hand, every one of the large shipyards would be closed to-day.

In passing from this doleful retrospect to the outlook of the present year, we regret that we cannot picture it in rosier colors. Recent bids opened for one battleship and two armored cruisers disclosed figures far below (about 10 per cent.) the estimates on the same work of last year, and are an index, first, of the needs of the shipbuilder for more contracts, and, second, that the economies of construction have been carefully observed at every point. The closing days of the year recorded the placing of orders for two coasters and a night bay steamer. The orders for ferryboats for several railroad companies with terminals at the port of New York have furnished the backbone of merchant contracts for the year. This class of boat has passed through a complete evolution within the last few years, and the double-end, double-screw, steel ferryboat is now accepted as standard.

Fortunately, the Naval Board of Construction has decided to adhere to its policy of recommending the building of two battleships and one armored cruiser, besides necessary attendant craft, each year. Thus it is proposed that three battleships be ordered by the coming congress, to make up the average fof the two years. Besides this, it is understood that five scout cruisers and several torpedo vessels are to be included in the bill. The scout cruisers are of the same general type as those now under consideration by the department, general features of which will be found elsewhere. The building of the Panama canal will necessitate vessels and barges of every class, from steamers suitable for the New York-Colon service to dredges and barges engaged in the actual construction work. Already plans are being prepared for the large steamers, and the amount of traffic, both freight and passenger, to and from the isthmus, is even now increasing at a very rapid rate. The extension of the coastwise laws to the Philippines will assuredly stimulate the growing trade with that archipelago, and especially the shipbuilding industry of the Pacific.

Be these influences as they may, the general trade conditions of the country are infinitely better than they were a year ago, and there must shortly be a demand for ships in the natural development of our inland and coastwise commerce. For any definite improvement in the foreign trade we must, however, await the decision of congress. At the time of writing, the bill prepared by the combined congressional commission to propose a plan for aiding the American merchant marine, is about to be presented, but comment must be deferred till the next issue. It is evident that the commission has worked with a view of drafting a law for the encouragement of the American ship that would not entail a direct charge upon the treasury. Just what action congress will take on this measure it is folly to predict.

The one bright spot in the year's work is the activity of the ship yards on the great lakes in preparing for the coming season. Many orders for new ships have been placed, and, while the number of ships and tonnage does not equal that of two years ago, the record year, it is most satisfactory, and includes several vessels of the largest displacement of any fresh-water ships afloat. The great lakes yards offer an illustration of how the cost of shipbuilding may be reduced when the demand is sufficient to warrant specializing the types of vessels.

TURNING now to the progress made in naval architecture and marine engineering, the stagnant condition of the industry naturally has prevented any marked developments. In good times, when the shipbuilder proposes a radical change, the prospective owner will give it more consideration than in the dull season, when the trade does not warrant investigating new types. Another record has been made for the largest ship in the world by the addition of the steamer *Baltic* to the transatlantic fleet, exceeding by 26 feet in length and about 3,000 tons, the registered dimension of any previously existing ship. The increased earning capacity of these huge moderate speed vessels has been very marked, and the year has recorded several orders for this type built in Great Britain, practically all for the North Atlantic lines, and one measuring over 700 feet is now under construction in Germany.

The development of the cargo steamer has received a great deal of attention, especially the means of loading and discharging bulk cargoes. The low figure at which thousands of tons of freight is carried has necessitated many radical changes in the plans and equipment of the ship. On this side of the Atlantic the latest successful production has been the *Augustus B. Wolvin* of the great lakes, which represents the most approved design for discharging by machinery practically the entire cargo. An unobstructed hopper, extending from the machinery space forward, a distance of some 400 feet, is built with slanting sides inside the ship. The unloading machinery, erected upon the pier, has free play through the large hatches, spaced 12 feet centers, and for supporting the deck a special arched girder is substituted for the usual deck beam. The success of this arrangement is so complete that many new vessels of the same design are now under construction. On the other hand there has been perfected a type of carrier which is entirely independent of apparatus on shore for discharging the cargo. Twin masts or derricks are spaced between hatches at frequent intervals, and are so rigged with booms, buckets and winches that they can unload in a very short time. Another type of vessel which has met with great favor and for which many orders were placed last year is the suction dredge. The principle on which it acts was practically undemonstrated ten years ago, and by its adoption the capacity of discharge per horse power has been increased almost tenfold over the bucket type. The national work of deepening our harbors and navigable rivers has now assumed such large proportions, and the equipment demanded by the construction of the Isthmian and New York state canals, will call for many more of these vessels. Mention should also be made of the turret steamer, which has met with as much favor on the other side as the whaleback type has with disfavor in this country.

During the year, probably the greater attention has been given to ship propulsion than to any other phase of marine engineering. The year has marked the check in the tendency of increasing steam pressures. From the time of the old beam engine, of forty years ago, using 30 pounds per square inch, there has until the present been a steady rise in pressure, corresponding to the increase in strength of materials and engineering progress. The naturally increased weight attending higher pressures, the difficulty of keeping steam-tight joints and other reasons, have caused the engineer to look elsewhere in the cycle for further increase of efficiency, and the steam turbine has, it is interesting to note, shown him that this increase may be obtained at the lower end of the expansion, as well as by the use of superheat. The progress made with the steam turbine has been a natural one along the lines indicated by former installations. Increased vacuum is now obtained by means of dry air pump, or the augmenter condenser, while the resulting lower temperature of the hot well is compensated for by the multiple feed water heater.

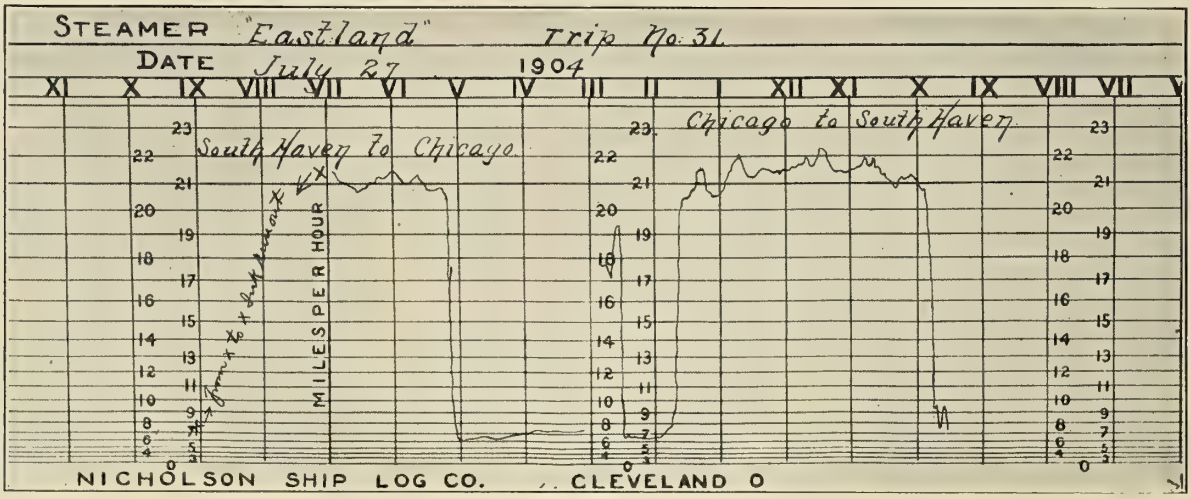
By far the greatest interest is directed to the two 23-knot express steamships building for the Cunard line, and which are to be powered with steam turbines. The coming year will doubtless witness the trials of several Atlantic steamers, now nearing completion, with this type of propelling machinery, and these results must be the real index as to the commercial efficiency of the Parsons steam turbine for vessels of average speeds. We have from time to time given account in these columns of the progress of this class of work, and it is well known that the advisory board, which has the design of the Cunarders in hand, and which is composed of the most eminent British engineers, has been conducting exhaustive experiments on model hulls and propellers. The results of the four new Irish channel vessels, all of the same dimensions, two of which are fitted with turbines, appear to indicate from the preliminary figures given out that the turbine is most successful.

The recommendation of the Admiralty Boiler Committee to henceforth adopt the water tube boiler in all British warships, has done much towards clearing up the controversy and defining the efficiency limits of the two classes. However, the old Scotch boiler with all its foes will be seen for many years to come in most classes of merchant ship. In this respect it is interesting to note that the Cunard vessels are to be fitted with this class of boilers. Assisted draft, thereby increasing the evaporation per square foot, has prevented the abandoning of the Scotch where large power is required.

All things considered, the past year may be looked upon as an important one in partially demonstrating the theories advanced during the last few years, and we may look upon the present year as one for further demonstrating whether or not our investigations have been molded along correct lines.

High Speed of the Eastland.

The steamer *Eastland,* built by the Jenks Shipbuilding Company, in 1903, for the Michigan Steamship Company, has proven to be a very fast and successful boat. The speed attained is strikingly illustrated by the record here reproduced from the Nicholson ship log, which is fitted on the steamer. On July 27 the run from Chicago to South Haven, a distance of 79.5 miles, was made in 3 hours 44 minutes, and the average speed for 3 1-2 hours of the run was about 21 1-2 miles. The variations in speed at starting and during the trip, due to the wind and variations of engine revolutions, are clearly indicated on the card.


STEAMER "Eastland" Trip No. 31.
DATE July 27 1904
South Haven to Chicago.
Chicago to South Haven.
MILES PER HOUR
NICHOLSON SHIP LOG CO. CLEVELAND O

CONTINUOUS LOG TAKEN ON THE FAST RUN OF THE EASTLAND.

The *Eastland* is 275 feet long, 38 feet beam, and 22 feet 8 inches deep, molded. There are two sets of triple-expansion engines, the diameters of the cylinders being 21, 34, and 56 inches respectively by 30 inches stroke. Steam is furnished by four Scotch boilers, 13 feet 6 inches diameter by 12 feet long.

Serious Collision of the Martello.

The steamship *Martello* was prevented from foundering in a head-on collision with the *Holywell,* on the east coast of England, by the fact that the second bulkhead was well forward. As shown in the engraving taken of the ship in dry dock, the forward collision bulkhead was entirely destroyed, and if it had not been that a second bulkhead was well forward it is probable that the ship would have foundered. The collision took place at 2 o'clock in the morning of February 21, 1903, while the *Holywell,* hailing from Sunderland, was on a voyage from Middlesbrough via London for the east coast of Africa. The *Martello* sailed from New York to Hull. The *Martello* struck the *Holywell* on the starboard side amidship, and sank her in fifty minutes.

Before the vessel was built, when the plans were presented to the owner, he passed the remark that if the vessel at any time should have a big collision the second bulkhead would be of little use, so the bulkhead was accordingly built 40 feet forward.

FAST PASSENGER STEAMER EASTLAND, IN SERVICE ON LAKE MICHIGAN.

It is a remarkable thing that after nineteen years she should be saved in reality as a result of his argument.

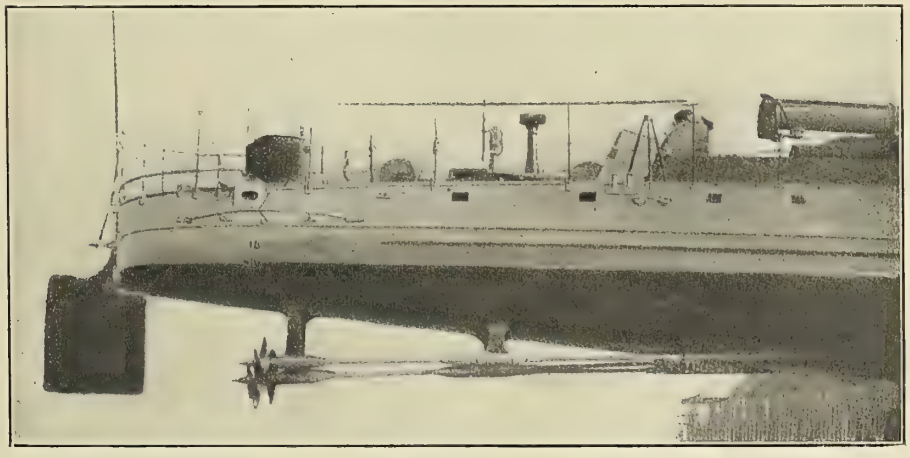
Only one life was lost through the collision—that of a fireman who lay asleep in his bunk. The *Martello* was eventually repaired at Hull and is now trading between the United States and Hull. She is 370 feet long, 43 feet 1 inch beam, and 28 feet 4 inches deep. The propelling machinery consists of one set of triple-expansion engines, 26 1-2, 50, and 82 inches by 57 inches stroke. She was built by the Earles Shipbuilding and Engineering Company, Ltd., of Hull, England, in 1884, and in 1900 was fitted with new Babcock and Wilcox water-tube boilers. The vessel belongs to Thomas Wilson, Sons & Company, Ltd., Hull, England.

French Turbine-Propelled Torpedo Boat.

The first vessel of the French navy to be fitted with Parsons steam turbines has recently completed her trials, and is described in London *Engineering*, from which the accompanying facts and photograph are taken. The boat is numbered 293, and is 130 feet long, 114 feet beam and 94.6 tons displacement, with 19.5 tons load—in fact, is as near like the reciprocating engine-propelled torpedo boats of the same type as was possible to build, modified only to suit the turbine machinery. Two Normand water-tube boilers supply steam at 250 pounds pressure. The hull was built by Augustus Normand & Company, and the engines by Parsons Marine Steam Turbine Company. A high-pressure turbine is fitted on the port wing shaft, and exhausts into the intermediate-pressure, driving the starboard wing shaft, which in turn exhausts into the low-pressure turbine on the center shaft. This arrangement is only for speed above 17 knots; for all lower speeds an additional high-pressure turbine is employed, and the ordinary high-pressure which receives its exhaust becomes the first intermediate. This cruising turbine, as it is called, is placed on the center shaft and coupled to the low-pressure turbine, and, when not required, steam is simply shut off from it and it revolves idly in a vacuum. The auxiliaries include two Normand condensers, Weir main and auxiliary feed pumps, air pumps driven by a reduction gearing from center turbine shaft, Norman feed heater and filter, etc. Reversing is accomplished by a set of reversing blades fitted at the aft end of the low-pressure turbine.

THE DAMAGED MARTELLO DISCHARGING HER CARGO.

THE MARTELLO IN DRY-DOCK, SHOWING DAMAGED COLLISION BULKHEAD.



FRENCH TORPEDO BOAT 293, THE FIRST TO BE FITTED WITH PARSONS TURBINES.

The eight-hour official consumption trial at 14 knots resulted in boilers taking 683 pounds of coal per hour, and the two hours' continuous fuel speed runs gave an average of 26.205 knots and 26.663 maximum. An analysis of the trials compares favorably with reciprocating-engined torpedo boats, for with the cruising turbine low coal consumption was attained such as not heretofore attained with turbines. In some torpedo boats the reciprocating engine has been fitted for cruising speeds with turbines for full power, which double arrangement is not altogether to be recommended.

An Engineer's Yarn.

BY WILLIAM J. AUTEN.

We were well by Hatteras on the trip north, bound for New York, when I came on watch at 12 o'clock one evening last February. Although we only made about 10 knots, loaded as deep as we were, still, even by moderate reckoning, we should make the Highlands by at least 4 bells the next afternoon, which in turn ought to leave us tied up at the dock that same night.

Possibly you who do not "go down to the sea in ships" cannot realize how much a lone coaster longs for a night ashore, especially when his packet hails from Boston, with but an occasional call at the Imperial city, and, too, when his hope of future happiness is enthroned in one of Gotham's "double-deckers"; despite his glowing descriptions of his native city of such social standing that one of its tea parties has been handed down through history.

We carried the usual complement of a 2,000-ton freight steamer —a fair-sized triple, two Scotch boilers, with Howden draft and a *leaky* condenser, over all of which your humble was the usual grumbling, second-assistant engineer.

Now, to be called out at midnight to go on watch, doesn't improve a fellow's temper, anyway, and when I found we were only doing about 47 "turns," carrying 132 steam and fans shut down, I naturally felt grumpier than usual, for the picture of those glimmering lights at the pierhead were growing fainter and fainter. A glance at the log slate confirmed the worst—the turns had dropped from the first watch at 88.2 to 58.6 in the watch of our glorious third; the steam from 184 (she popped at 185) to where the pointer bobbed now, and the wind from 3 1-2 inches to 0. The remarks column settled it. No Pier 2 to-morrow. It ran:

"9:02 P.M. Blowed starboard boiler down; 2-inch surface below.

"9:46. Ditto for port boiler.

"10:02. Shut down blowers.

"10:12. Slowed engine to 50 turns.

"11:04 and 11:50. More blows.

"Condenser leaking badly, boilers priming, and drains wide open."

I closed the slate and stumbled out into the fire room. Plenty of water—over two gauges in each boiler. A predominance of salt sickles around trifling leaks bore mute evidence to the fact that the condenser was leaking. I relieved the "third." He didn't need any second invitation, either.

After promenading the lower grating awhile, dodging the showers from the high-pressure rod as she "took it over" (that metallic packing never would hold water, anyhow), I thought I'd make a last feeble attempt to drive her through the water a little faster, and tapped the throttle down a bit, but she wouldn't stand it; a couple of reports from the cylinder artillery were very convincing.

And then? Well, I lapsed into one of those day dreams that last the whole watch, including its usual duties, but which is made up of a certain pretty shore vision. But, strange as it may seem, an idea disturbed the vision that night; and it grew, too, as the fancy receded, and finally enthroned itself in burning letters, as my little fairy vanished with a longing adieu. Those letters sputtered and burned, and finally spelled themselves into "Superheated Steam."

My promenade down the grating came to a sudden pause, and a few seconds later I started off with a definite idea in view. The feed pump was the objective point. This I slowed down and watched the valuable feed water run into the bilge; but it was pretty brackish, anyhow, so I didn't care. The water came down in the boilers all right, but I kept just an inch or so in sight before I let the pump go again. Dangerous? Well, perhaps, but I had my eye on it, and we were bound for New York now.

It is part of the usual routine to wake dago firemen out of a trance, but they came out of theirs quicker than usual that night. The fans were going, too, before the pointer had gained more than a pound or two.

Then I went for the main stop, my trump card, shut it down full, and then opened it about a turn or so. The throttle got a dose of the same medicine next. I stood by, then, and watched results for a minute. She was doing 52 with less grumbling than her 45 a half-hour ago.

As the steam came up I opened the pass-overs. That did the business, I tell you. Even the oiler had to wake up, for my new supply of "hot air" was talking French to the speed counter—an agreeable language just about that time. I had to open the stop a bit more, for the boiler room gauge was creeping up too fast; however, she behaved like a lady, and only had one or two attacks of the sniffles—none very serious. I got the drains shut up, too, except the low, which I left open a bit to keep on the safe side.

The howl for steam became quite genuine; reminded me of the days of government trial trips, when the speed bonus was in vogue, with just two differences: one the size of the plant concerned, the second the size of the bonus, the former being as small as the latter was large. Whenever we made a pound or two of steam I would open the pass-overs a trifle more, then dive for the fire room, to help my dagoes do stunts with their slice bars before she dropped back again.

This was the state of things when the "first" showed up at 8 bells, but he only smiled when I told him what I had been up to. (He was a New Yorker himself, by the way). There was a rather serious look on his face, however, when I wrote four inches under "Air pressure in ash pits" on the log slate. We figured up the turns together and got 72.4 for the average. Then, after a talk about the water, which hugged the bottom of the glass pretty closely, he decided it was all right, and I departed for the land above decks, where the air is free and pure and you don't have to know the difference between a safety valve and a cylinder head.

That was a straggly looking trio that passed my state room on their way to the forecastle, and for once in their lives I believe those firemen earned all that the company paid them.

My system lasted all the way in. We crossed the bar at about 6 bells that afternoon, and were tied up shortly after dark. Then I made my vision a real fact.

Valve Diagram with Meyer Cut-Off.

In the October number of Marine Engineering an instance was given of how the indicator may be used to take a diagram of the motion of a valve, where the travel does not exceed the length of the drum. This method of analyzing a valve motion deserves to be better known, as it proves highly instructive to the engineer who applies it to his own engine.

Here is reproduced a section of the valves and a pair of diagrams taken on board the steamer *G. W. C*——, from a Meyer cut-off gear of 4 inches travel of main and cut-off valves. A drum made from a piece of 5-inch brass tubing, and run from the regular reducing motion gave a card 11 inches long. The large ellipse gives the motion of the main valve, as was explained in the former article. The narrow diagonal diagram represents the cut-off valve. Theoretically, this should be a straight diagonal line, since the center of the eccentric is opposite to the crank center. But the regularity of the connecting rod causes it to sag in the middle of the stroke; and the stretch in the cord is responsible for the double line instead of a single one.

To show on the diagram the different adjustments of the valve, this line must be reproduced in different heights, corresponding to the amounts the two valve plates are moved apart. Steam is cut off by these plates at the outer edge of the steam ports *through* the main valve. The inner edge of the same port controls the opening to the valve seat port, and is represented on the diagram by the periphery of the ellipse. Accordingly, the part through the main valve will be located along the side in the ellipse

and will show the opening in reference to the cut-off valve, as the steam port in seat shows the opening given by the main valve.

With this arrangement it will be noticed that the steam opening overlaps slightly into the exhaust opening *in the diagram,* but this need cause no confusion.

On the diagrams here given can be seen the earliest and latest points of cut-off possible in this particular case, while the steam opening used under ordinary running conditions has been shaded, for easier comparison with the cylinder cards taken under the same conditions. This grade of expansion is a little beyond the range of the extra cut-off, so that the steam is following until the main valve shuts the port at 32 inches. The common trouble with a plain slide-valve engine of getting the heaviest M. E. P. on the down stroke has been overcome in this case by raising the cut-off valve (lengthening the rod). The result is an interesting difference of appearance of the top and bottom cards.

One should expect a more rapid cut-off where an independent valve is provided. It is revealed here, however, that it is pretty slow at any point, and especially after passing the middle of the stroke, when the motions of the two valves nearly coincide. Giving the cut-off valve longer travel would improve the gear in this respect.

In cases where changes in the valve gear are proposed, such diagrams as these would be useful in studying the effects the changes would have on the steam distribution. Thus, for instance, difference between top and bottom lead is allowed for by raising or lowering center line and other lines half the amount of the difference. Travel is changed by regulating vertical height of ellipse; the laps, by drawing new lines. Advancing eccentric on shaft tends to make the ellipse narrower, until it would approach a diagonal line, as we have seen in the case of the cut-off eccentric, whose center is opposite to that of the crank.

It is interesting, also, to note that the effect of linking up the main valve (shortening its stroke) is to give a later cut-off.

O. O.

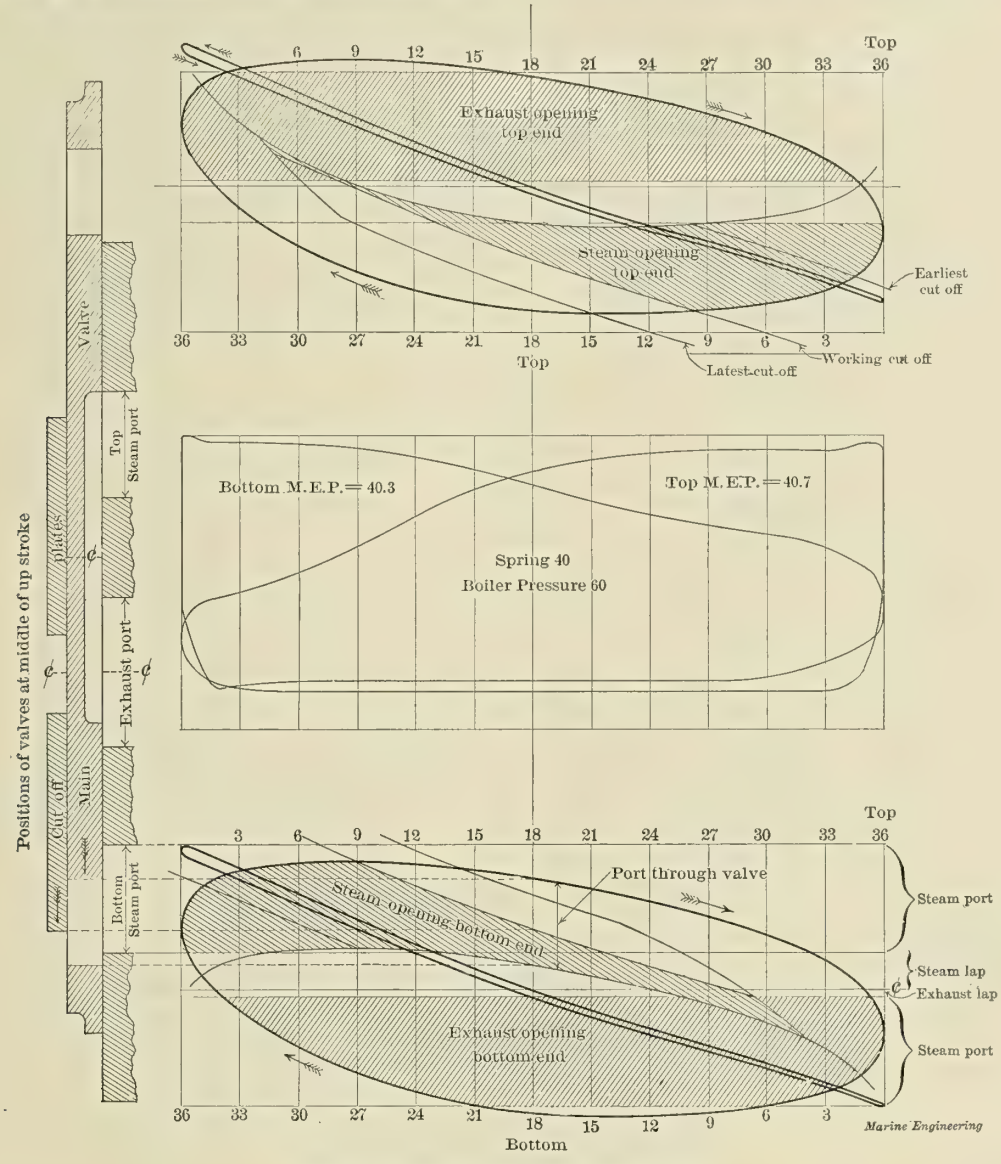


DIAGRAM OF VALVE WITH MEYER CUT-OFF GEAR.

ENGINEERING SPECIALTIES.

New Vertical High-Speed Engine.

The illustrations presented herewith show a small vertical engine, recently placed upon the market, which possesses unique features of great merit. From Fig. 1 it will be seen that the engine has easy curves and graceful lines, giving it a handsome appearance. It is fully enclosed, preventing the foreign matter getting into the working parts or the oil from leaking out, yet the parts are easily accessible upon the removal of the enclosing plates by simply turning a milled hand nut and lifting out.

The most striking feature of the engine is its oiling system. Up to the present time most of the small engines upon the market have depended almost entirely on the splash system for lubrication, which is seldom thoroughly effective, and for vertical double reciprocating engines is positively unreliable. The system

embodied in this engine is claimed to be different from any of the usual methods, and has proven very successful. Every frictional surface is running on oil, with the effect of almost eliminating wear. Positive circulation of the oil is secured by means of an oil pump *A*, driven from the main shaft by the eccentric, *K*, shown in Fig. 2. The oil is forced up through the tube, *B*, into the small strainer, *C*, from whence it drops into an oil box through the bottom of which four tubes project. In one side of each of these tubes there is a slot, so that when the oil box contains only a small amount of oil, each tube can take only its proper proportion. In case the box is full, all the tubes get more oil in proportion to the increased height.

Two of these tubes, *F* and *C*, and two on the opposite side not lettered, apply oil to the guides, the oil dropping into a small oil trough through, *G*, from which it runs into the bearing through a small oil hole. The crosshead pin is supplied by the tube marked *E;* the oil drops into the cup, *H*, and fills the cavity between the bolt and inside of crosshead pin and the oil grooves. The oil dropping from the crosshead is caught in two pans attached to the inside of the covers. From these oil pans it runs down the inside of the cover dropping into a cup in the top of the main bearing cap. Instead of oil grooves at the top and bottom of the main bearing, as ordinarily made, in this system the oil is supplied at the sides where the bearing is cut away at the joint. When the strain from the connecting rod is up the oil is carried to the bottom of the bearing, but when the load is reversed there are no oil grooves to carry away this oil. The crank pin is oiled through the tube marked *D*, which discharges into a crank oil ring inside the eccentric, which in turn discharges into the crank-pin oil tube, and flows across the crank-pin bearing. The crank-pin oil ring in addition to its independent supply catches the drip from one end of the main bearing. The eccentric is well oiled by the drip which it catches from the other end. A portion of the oil as it drops back into the bottom of the frame drops upon an oil filter where it is thoroughly cleaned and purified. The large base gives the oil an excellent opportunity to cool and settle.

In their own shops, where the manufacturers have been experimenting with this engine for some two years past, it is stated that an engine adjusted and filled with oil on March 10, had no adjustments made of any character up to July 15, and no oil added except to fill the sight feed lubricator. It has since been operating from fourteen to sixteen hours per day driving a blower, and after a lapse of over four months really needed no adjustments or fresh lubricant, running almost as noiselessly as at first. It will thus be seen that this engine is particularly valuable for applications where continuous operation is required or peculiarly heavy duty imposed upon the engine, as in driving blowers, dynamos, pumps, etc.

Further information can be obtained from the designers and builders, American Blower Company, Detroit.

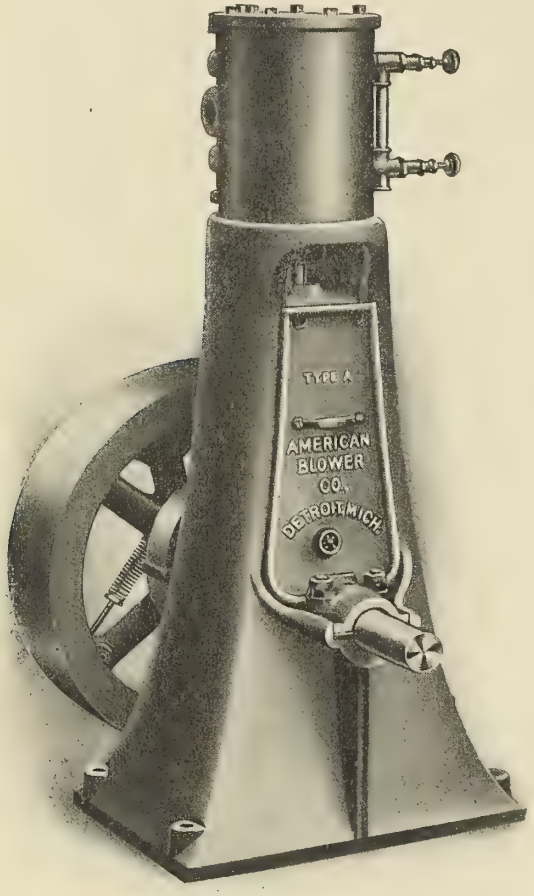


FIG. 1.—NEW VERTICAL HIGH-SPEED ENGINE.

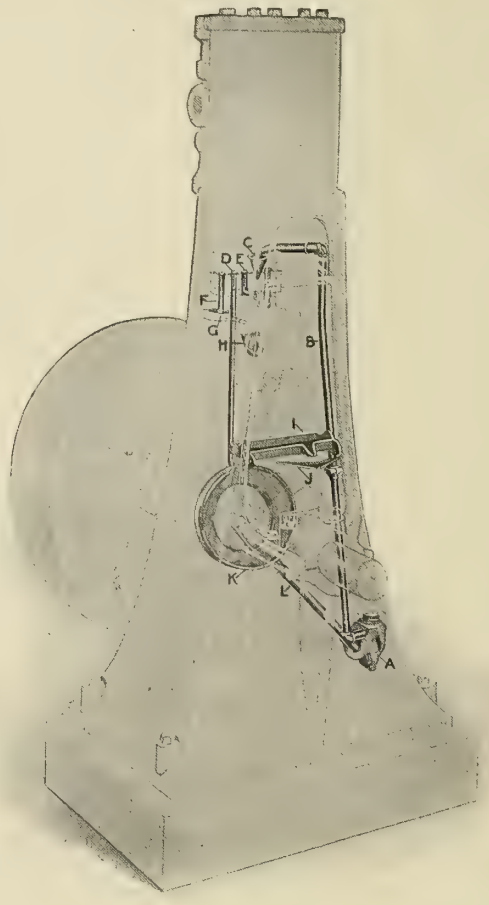
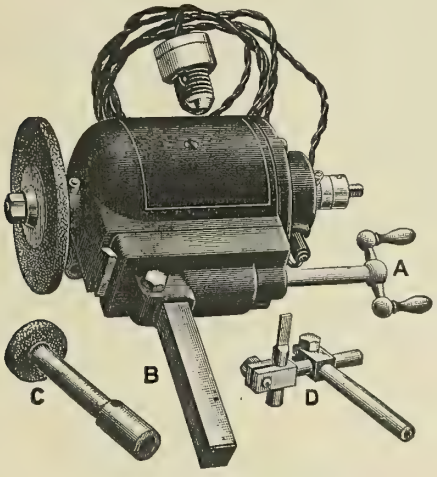


FIG. 2.—POSITIVE LUBRICATING SYSTEM.

Portable Electrical-Driven Grinder.

An electrical-driven grinder, recently placed on the market and here illustrated, has a wide range of work, such as grinding centers, cutters, reamers, dies, and for surface, parallel, and internal grinding jobs of all kinds. The machine is portable, the largest size, of 1 horse power, weighing but 65 pounds; and the smaller size, of 1-4 horse power, weighing only 16 pounds. The grinder is set in the tool post of a lathe, planer, or milling machine, or may be clamped in a vise. The equipment includes all the parts shown in the cut, and the machine is started running

by simply attaching the plug to an incandescent lamp socket and turning on the current. The standard motors for driving the grinders are wound for either 110 or 220 volts, direct current unless otherwise specified. All the parts pertaining to the motor are incased in a dust-proof shell.



PORTABLE ELECTRIC-DRIVEN GRINDER.

The spindle carries taper cone bushings, running in 3-degree and 45-degree bearings. They are adjustable to wear, by means of a nut on the rear end of spindle, and are provided with dust-proof caps. The V slide has a 3-inch travel, by means of a worm through handle *A*, and is fitted with gib to take up the wear.

The shank, *B*, is of steel, and is fitted to hole in V cap. It is held in position with a screw, so that different size shanks can be used. The extension mandrel, *C*, is used for internal grinding by removing the regular wheel and attaching mandrel in its place.

The tooth rest, *D*, is a valuable attachment, serving as an index for cutter and reamer grinding.

As readily seen this combination grinder is a time-saver, but simple in its adjustment and easy to operate. This grinder is manufactured by the Hisey-Wolf Machine Company, Cincinnati, Ohio.

The Sensible Screw Driver.

The *Sensible* screw driver, made by the Vanderbeek Tool Works, Hartford, Conn., is built on an entirely new principle. It is admitted that the only working point of a screw driver is the extreme end; therefore, if an inserted blade is used, it may be made of the best steel manufactured at a minimum cost. It may also be hardened and brought to a positive spring temper which will be guaranteed against breaking or twisting. In this *Sensible* screw driver the blades slide in the slot and are held by friction that will not get out of order. Each blade is double-ended, thus giving two sizes of drivers to each blade. The extra blades are carried in the recess in the handle, and here again is a unique feature of this screw driver. The cap to the recess is of tempered steel and forms a substantial head for hammering on. Of course a screw driver is not intended to be used as a chisel, but at some time every one does make this use of it. The *Sensible* screw driver can stand this treatment without damage. It is made in five sizes and each size is supplied with a strong steel snap for holding the screws, as shown in the illustration.

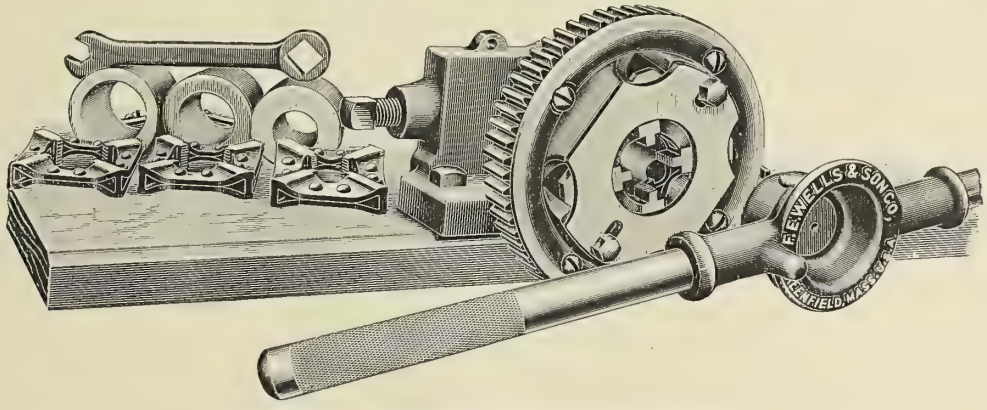
New Pipe-Threading Machine.

F. R. Wells & Son Company, 37 Biddell street, Greenfield, Mass., is putting on the market a new pipe-threading machine, as shown herewith, for threading 1 to 2-inch pipe. It is a very simple, compact little machine weighing about 45 pounds, that is used much like a hand die stock, only having gears to multiply the power so that one man can alone easily thread pipe that usually takes two with a hand stock.

The die is held in the large gear, which has a threaded shank that screws into the main frame and acts as a lead screw to start

SCREW DRIVER WITH REMOVABLE BLADES.

dies. The pipe is centered by means of bushings the same as in an ordinary die stock, and is held from turning by two vise jaws on the back of machine operated by set screws. The gears are all machine cut and all castings are of malleable iron to save weight and give ample strength even for the hardest usage. The



PIPE THREADING MACHINE FOR FROM ONE TO TWO-INCH PIPE.

die can be bolted to either a bench or post and the vise, which is a part of the machine, can be used in place of an ordinary pipe vise. The *Economy* dies, made by the same company, are furnished with the machine, but any standard solid square pipe die will fit.

The moderate weight and small amount of work permits the machine being sold at a low price.

ABSTRACTS FROM RULES AND REGULATIONS OF LLOYD'S REGISTER OF SHIPPING.

EDITION OF 1903.

ARRANGED BY HARRISON S. TAFT.

Among the large number of hand books published upon engineering subjects, there does not appear to be one devoted especially to the subject of shipbuilding, covering that part of the work which distinctly comes within the supervision of the hull department. Most of the hand books now in existence generally deal with the engine side of the shipbuilding industry, devoting but a few pages or so to the questions of vital interest to the hull department, and these few dealing more on theoretical matters than on the practical details of steel construction as applied to shipbuilding.

In the rules and regulations of the various shipbuilding societies, as *Lloyd's Register,* the *American Bureau of Shipping,* etc., we find the details of practical shipbuilding fully and laboriously treated, but somewhat involved and beyond the limits of an ever-ready hand book. Consequently, it is believed that there is room among the archives of shipbuilding literature for a series of articles treating entirely on the practical side of the subject in such a way as to be equally within the comprehension of the American ship fitter, as well as that of the executive heads of our shipyards.

In the following articles the endeavor has been made to give a thorough and accurate abstract of the rules and regulations of the principal shipbuilding societies, of the Bureau of Steam Navigation, and of certain information published by our Navy department. The subject matter is set forth in tabulated form wherever possible. It is assumed that the reader has some knowledge of the original set of rules and regulations from which the abstracts have been made, so as to readily understand the terms and expressions used.

1. Measurements.

TABLE I.—MEASUREMENTS FOR SCANTLING NUMBERS.

ITEM.		One-Two-Three-Deck Vessels.	Spar Deck Vessels.	Awning Deck Vessels.
Length.—From aft part of stem to fore part of stern post	L	On upper deck.	On upper deck.	On main deck.
Depth and girth	D-G	To " "	To main "	To " "
Breadth	B	Maximum beam.	Maximum beam.	Maximum beam.
First Number.—Frames, reverse bars, floors, bulkheads	F	a. $\frac{1}{4}B+D+\frac{1}{4}G$	$\frac{1}{4}B+D+\frac{1}{4}G$	$\frac{1}{4}B+D+\frac{1}{4}G$
Second Number.—Keel, stem, and steel frame, keelson and stringes bars, keelson stringers, shell, and deck plates	S	$F\times L$	$F\times L$	$F\times L$
Equipment Number.—Anchors, chains, warps	E	S	b. $F\times L$	Complete, $S+\frac{1}{8}S$ Partial, $S+\frac{1}{L}\frac{1}{8}S$

a. Three-decker only. $\frac{1}{4}G$ = girth of $\frac{1}{4}$ midship section. Depth to include round up of beam.
b. Depth and girth measured to upper deck.
c. Partial awning decked, poop, top gallant fo'castle, bridge house or a raised quarter-deck vessel to have their equipment number equal to that for a flush or spar-deck vessel increased by that proportion of the addition made for a complete awning deck, which the combined length of the erections bear to the length of the vessel.
Sailing vessels with a poop, top-gallant fo'castle, or raised quarter-deck to have their equipment number measured by $\frac{1}{15}$ beyond that which it would be if of flush-deck type.
In spar-deck vessels two depths may be taken off actual depth to main deck in determining stringer plates, steel deck, and tie plates.

2. Stems, Stern Frames, Bar Keels.

The lengths of scarphs should be at least 9 times thickness of keel.

3. Stems.

STEMS are to equal scantling of a bar keel and to maintain their full section to load W. L. with a reduction to 3-4 said section at head.

4. Stern Frames.

1. The aft end of the scarph with keel should be at least 1 1-2 frame spaces forward of the stern post in sailing vessels and paddle-wheel steamers, and 2 1-2 frame spaces forward of propeller post in screw vessels.

2. With screw vessels whose second number is 20,000 and over, the propeller post is to be extended so as to be connected to a deck beam with a deep transom plate.

3. When stern frame is made in more than one piece, length of scarphs to be not less than three times, and width not less than 1 1-2 times the width of stern post, secured by at least four rows of rivets.

4. Rudder posts of sailing vessels and of paddle wheel steamers may be reduced from full section at bottom of transom frame to 3-4 said section at head of post. In single screw steamers the reduction to be according to rule.

5. GUDGEONS to be spaced from 4 to 5 1-2 feet apart. In depth to be not less than 7-10 of the diameter of rudder head and in thickness 1-2 diameter of pintles.

5. Keels and Center Line Keelsons.

1. CENTER THROUGH PLATE KEEL AND KEELSON.

a. These shall extend from lower edge of keel to top of floors, with side plates of same depth as a bar keel. The combined thickness of center and side plates shall equal that of a bar keel. There shall be horizontal plate stringer, at least 3-4 breadth of garboards in width, on top of floors at middle line of ship, and connected to top of vertical keelson plate with continuous double angles on under side of said floor plate.

When the second number is 15,000 and under 18,000, a vertical bulb plate of same depth as main deck beams is to be riveted between two continuous angles on top of said horizontal plate stringer at middle line of ship.

b. When the continuous center keelson extends above the floors, there is to be a horizontal plate stringer on top of floors on each side of vertical keelson, each being in width 1-3 breadth of garboard, and connected to said keelson with continuous fore-aft angles on upper side of said horizontal stringer plate.

When the second number is 18,000 and above, the center keelson must extend above the floors a sufficient height to allow of two top angles with a rider plate, besides the two angles to be secured to the top of horizontal stringer on top of floors.

2. MIDDLE LINE INTERCOSTAL KEELSON.

a. These are to be worked level with top of floors with a bulb plate, 2 inches more in depth than main deck beam, let down below top of floors and riveted to top edge of intercostal plates and embraced by two continuous angles on top of floors. The letting down of the bulb plate may be dispensed with if the intercostal plates are worked high enough above floors to be embraced by the said two floor angles.

b. When used with hanging keels and second number is 13,000 and under 18,000 a vertical keelson plate with double top bars and a rider plate, riveted bewteen the two continuous angles on top of floors is to be worked in lieu of said bulb plate; the depth of vertical keelson to be sufficient to afford a proper working of angles.

When said number is 18,000 or above, said depth to be 3-4 of a specified height.

3. MIDDLE LINE KEELSON ON TOP OF FLOORS.

They are to be worked with double angles and a rider plate on top and connected to floors between double continuous fore-aft angle bars riveted to top of floors embracing said keelson plate. Rider plate to extend at least 3-4 length amidships.

When second number is 33,000 and above, a horizontal stringer plate not less than 18 inches wide must be worked on top of floors under said vertical keelson.

4. FLAT KEEL PLATE WITH MIDDLE LINE KEELSON.

a. When second number is or above 26,000, flat keel plates are to be worked double for 1-2 length amidships. They are to be fitted with an intercostal or a continuous plate keelson.

b. The middle line keelson must be secured to flat keel plates with continuous double angles for a continuous keelson or with intercostal angle clips for an intercostal keelson, and be secured to floors with double clip equal to reverse bars. The top of a continuous keelson to be worked as under 5-1; that of an intercostal keelson as under 5-2.

c. When second number is 13,000 and under 15,000, or when length of vessel is over 10 depths, the top of said keelson must be fitted with double top angles and a rider plate as above. When said number is 15,000 and over, the keelson is to be worked 3-4 of a specified height above floors as above.

6. Frames.

1. At extreme ends of ship lower part of opposite frame angles should be lapped and riveted to each other. Scarph pieces when used at center line to have butts at least 3 feet long and to extend for 3-4 length amidships. Cant frames to be spaced at knuckle not more than the regular frame spacing of ship.

2. In full form vessels frames at forward end between collision bulkhead and 3-5 length should be doubled to turn of bilge; but to margin plate in double bottom vessels.

3. Vertical flange of frame angle on solid floor frames in double bottom vessels may be of same size as horizontal flange of said frame.

7. Floor Plates.

1. For 1-4 length amidship they should extend up the bilges, on outside of frame, above top of keel not less than two times depth of midship floors. To be not less than 1-2 midship depth at 3-4 of half breadth and not less than molding of frames at ends. To be secured to all continuous center line keelson plates with double clips equal to reverse bars.

2. Forward and aft of 1-4 length amidship, the height of floor at bilge may be reduced until floors are straight on top. At ends of ship the center depth is to be gradually increased so as to securely connect the sides. No floor plate is to be of less depth on center line than midship floors.

3. When thickness of midship floors is 9-20 of an inch and above the floor for 1-10 of vessel's length fore-and-aft 3-5 length midship may be 1-20 of an inch less in thickness, and those at extreme ends 2-20 of an inch less in thickness than midship floors.

4. Bulkhead floor plates to be made deeper than other floors to admit of bulkhead plates being riveted to floors above reverse bars.

5. Boiler room floor plates to be 4 pounds heavier, engine room floor plates 2 pounds heavier than midship floor plates.

6. Limber holes to be cut in floors above frames so as to provide a free flow for water.

7. TRANSOM FLOOR to be of same weight, not less at center line than 1 1-2 times depth of midship floors, and to be secured to stern post with double clips equal to frame angles.

8. Reverse Bars.

1. To be fitted to every floor.

2. In topgallant forecastles when second number is 18,000 or above, reverse bars are to extend on alternate frames to said deck, or a double angle stringer is to be worked amid space.

3. Reverse bars to be worked double on every floor in engine and boiler spaces, extending at least from bilge to bilge. When the vessel is or above 17 feet in depth to hold beams, or where second number is 15,000 or above, they are to extend to stringer at upper part of bilge so as to be riveted to same.

4. When second number is or above 20,000, the reverse bars abaft aft peak bulkhead should all extend to upper, spar, or awning deck.

TABLE 2.—EXTENT OF FRAMES AND REVERSE FRAMES.

Type of Ship.	Frames.	Reverse Frames.
One deck vessel	All to upper deck	First number below 45, to upper part of bilges. " " 45 under 57 to U D and S S alternately. Exeept with hold beams, then to U D and top of H B S A alternately.
Two " "	" " " "	
Three " "	" " " "	To upper and main decks on alternate frames.
Spar " "	" " spar "	" spar " " " "
Awning deck vessel	" " awning " or to bottom of rounded gunwale	All to main deck.
Poop, fo'castle, bridge house decks	" " said deck " " " "	
Raised quarter, sunken fo'castle	All to raised quarter and forcastle deck stringers or to bottom of rounded gunwale	To raised quarter and fo'castle decks on alternate frames.
Sailing vessels	All to upper deck	First number 75 or over. All to upper deck.

Frames to maintain midship scantling for $\frac{3}{5}$ length. Reverse bars throughout. All frames W T Bulkhead to be worked double.
U D = upper deck. S S = side stringer. H B S A = hold beam stringer angle.

9. Bilge, Floor Keelsons, and Side Stringers.

1. BILGE KEELSONS FOR ALL SHIPS.

To be of double angles, riveted back to back, worked at lower turn of bilges. When depth of vessel is 15 1-2 feet or above, a similar keelson to be placed at upper turn of bilges.

2. FLOOR KEELSONS FOR ALL SHIPS.

a. When the second number is 13,000 and under 15,000, a double angle bar floor keelson is to be worked midway between bilges and middle line keelson. When said number is 15,000 and above, intercostal plates are to be worked in connection with said floor keelson, clipped to floors and shell with at least a 3-inch by 3-inch angle clip; but to shell with 3 1-3-inch by 3 1-3-inch clip, if said number is or above 21,700.

b. When size of ship does not require said intercostals, wash plate should be fitted between every floor at quarter beam for 1-2 length amidships.

c. Vessels of full form witnout inner bottoms should have an intercostal floor keelson between collision bulkhead and 3-5 length.

3. BILGE AND FLOOR KEELSONS AUGMENTATION.—SAILING SHIPS ONLY.

TABLE 3.—FLOOR AND BILGE KEELSONS.

	Steam Vessels.		Sailing Vessels.	
Second Number.	Floor.	Bilge.	Floor.	Bilge.
Under 13,000	.....	D A	.....	D A
13,000 under 15,0000	D A	D A	D A	D A
15,000 and up	D A + I R	D A	D A + I P	D A
24,000 under 27,000	.....	....	D A + I P + B P $\frac{3}{5}$ L	D A + I P $\frac{1}{5}$ L
27,000 " 30,000	.....	....	[1] D A + I P + K P T A R P $\frac{3}{5}$ L	D A + I P $\frac{1}{5}$ L + B P $\frac{3}{5}$ L
30,000 " 33,000	.....	....	[2] D A + I P + K P T A R P $\frac{3}{5}$ L	D A + I P $\frac{3}{5}$ L + B P $\frac{3}{5}$ L

K P = $\frac{1}{5}$ width middle line keelson. [2] K P = $\frac{1}{5}$ said width.
D A = double angles; I P = intercostal plate; B P = bulb plate; K P = keelson plate; T A = top angles; R P = rider plate.
K P T A R P = keelson plate with top angles and a rider plate; plus sign (+) means *with*; L = length amidship.
Example D A + I P + B P $\frac{3}{5}$ L means; double angle bar stringer with intercostal plates, and a bulb plate for $\frac{3}{5}$ length amidship.

4. SIDE STRINGERS.

a. All vessels with a single beam tier and with second number under 7,200, to have a double angle bar side stringer worked midway between bilges and said deck beams. But when said number is 7,200 and above and vessel is under 15 1-2 feet in depth, two such side stringers are to be worked between bilges and deck beams.

b. For number and arrangement of side stringers see Table 8.

5. When deck or stringer heights at sides is or exceeds 8 feet, additional traverse strength must be provided between stringers or decks.

6. Discontinuous side or bilge keelsons and side stringers should lap each other for at least 3 frame spaces.

10. Floor Clips.

They should be located on every floor throughout holds for all plate and angle bar keelsons and stringers.

11. Continuity of Keelsons.

All middle line, floor, and bilge keelsons should be worked continuously through all water tight bulkheads with water tight staple work.

12. Plate Side Stringers and Beam Stringers.

When worked without beams or when beam spacing exceeds two frame spaces the stringer plate is to have a knee bracket on alternate frames, clipped to shell, with a continuous angle riveted to face of reverse bar, or to reverse slips on frames. When beam spacing is 8 frames or above, stringers are to be connected to said beams with gusset plates, and have single face angles of keelson bar size; or double angles of specified size, with face plates as called for. A single face bar is to be provided in all cases when said stringer is worked without beams. Stringers are to be bracketed to all bulkheads when wide spacing of hold beams are used. When stringers are cut at water tight bulkheads they are to be bracketed to same; bracket plates 1-20 inch heavier than stringer plates, with double clips extending on bulkhead from inside of face bars on frame twice the width of stringer.

13. Web Frames in Engine and Boiler Rooms.

TABLE 4.—SPACING OF WEB FRAMES IN MACHINERY SPACE.

Second Number.	Spacing.
16,000 under 19,000...	At least 3 webs on each side in said space.
18,000 under 30,000...	8 to 10 feet apart.
30,000 and up.....	Not more than 8 feet apart.

They should be fitted to beam frames if possible; otherwise to have brackets above and below stringer plate in way of said webs. When hold beams are omitted in said spaces, said web frames are to be spaced closer than above.

14. Web Frames with Deep Side Stringers in Lieu of Hold Beams.

1. They should equal thickness of vertical flange of frame to which the webs are connected. Double angle bars equal to reverse bars are to be worked on faces of web frames and stringers. (Face bars may be worked single if of extra heavy scantling with double clip at diamond plate connections to side stringers). Web frames to be attached to double bottom margin plate with double clips. All side stringers to be worked intercostally and have double clips to webs.

TABLE 5.—SIZE OF DIAMOND PLATES.

Depth of web.....	14″	15″	over 15″
Size diamond plate.....	24″ × 18″	30″ × 21″	30″ × 24″

2. Side stringers 18 inches or above in width, with web frames more than 8 feet apart must have knee brackets to frames midway between webs.

TABLE 6.—WEB FRAMES AND STRINGERS IN LIEU OF HOLD BEAMS.

Center Depth to Lowest Tier Regular Beams.	Center Depth to middle deck 3-deck vessel.	Width of Web.	Spacing Frames.	No. of Stringers.	Notes.
Feet under 16	.....	14″	8	1	....
16 " 17	.....	15	8	1	....
17 " 18	Feet	15	8	2	Omit bilge stringer.
.....	17 under 18	15	6	2	Except awning deck, vessels omit bilge stringer.
18 " 21½	.....	15	6	2	....
.....	18 " 21½	18	6	2	Awning deck vessels 16″ web.
21½ " 22½	.....	15	6	3	....
21½ " 22½	.....	18	6	2	In double bottom vessels with margin bracket up bilge 3 floor depths.
.....	21½ " 22½	18	6	3	Awning deck vessels 16″ web.
22½ " 23½	.....	16	6	3	....
.....	22½ " 23½	18	6	3	Awning deck vessels 16″ web.
23½ " 24	.....	18	6	3	....
To Raised Quarter Deck, Feet.	24 " 25	16	5	3	4 hold beams and a W T bh. between aft engine room bh. and aft end of ship.
	25 " 26	16	4–5	3	
	26 " 27	18	4	3	
	27 " 28	18	4	4	

Depth taken from top of keel to top of first complete tier of beams other than wide-spaced beams.

15. Deep Framing for Vessels in Lieu of Web Frames.

1. Deep frame angle to have a 3-inch single riveted lap to regular frame angle. Knee brackets to lowest deck beams should be three times depth of beams. Side stringers should be bracketed to bulkheads. The size of brackets varying whether side stringers are worked continuously or are cut. Reverse bars should extend to upper deck in two-deck vessels; otherwise as stated in Section 8. Web frames in machinery space to be spaced about 10 frames apart.

TABLE 7.—NUMBER OF PLATE SIDE STRINGERS.

Depth to lowest tier of regular beams.....	Under 17′	17′ under 21½′	21½′ under 24½′
Number of plate side stringers.....	1	2	3

16. Deck and Hold Beams.

1. They should be located vertically over each other and as far as possible on frames which have their reverse bars carried up to decks.

2. All deck beams with a steel or wooden deck laid thereupon to be placed on alternate frames. When a steel deck, without wood decking, is less than 7-16 inches thick they should be placed on every frame. Short beams at hatches, engine, and boiler openings, should be fitted to every frame when no wood decking is laid on a steel deck.

3. Deck beams under a water-tight flat to be fitted on every frame.

4. Orlop or lower deck beams of sailing vessels to be 1 inch deeper than beams of upper deck of same length.

5. Through beams attached to deep web frames to be extra heavy when no steel deck is laid thereupon. Half beams to have large plate brackets to head of deep web frames.

TABLE 8.—SPACING OF DECK BEAMS AND TYPE OF HOLD STRINGERS.

Depths from top of keel to								Upper Deck.	Main or Middle Deck.		Lower Deck.		Orlop Deck.	
Class.	Upper Spar Awning.			Main on Middle.		Lower.		Beams.	Beams.	Stringers.	Beams.	Stringers.	Beams.	Stringers.
All....	Under 13*			....		....		Alt.	....	Double angles.	....	....	....	....
"	13	"	14	....		....		"	....	D A+B P 3/8 L D A+I P 5/8 L	....	....	....	....
"	14	"	15½	....		....		"	....	D A+12" I P+D F A	....	....	....	....
"	15½	"	16½	....		....		"	10 E	Plate	....	....	....	....
"	16½	"	17½	....		....		"	10 E	P S+S F A	....	....	....	....
"	17½	"	18½	....		....		"	10 E 2+4 Alt.	P S+S F A Plate.	.. .	....	....	....
"	18½	"	19½	....		....		"	10 E 2+4 Alt.	P S+D F A 3½"×3½"+F P Plate.	....	....	....	....
Sail...	19½	"	23	....		....		"	Alt.	Plate.	....	....	....	....
"	23	"	24	16 under	17	....		"	"	"	....	Double angle.	....	Double angle.
"	24	"	26	17 "	18	....		"	"	"	....	D A+B P	....	D A+B P
"	26	"	27	18 "	19	....		"	"	"	....	D A+B P+I P	....	D A+B P+I P
"	27	"	28½	19 "	20½	....		"	"	"	....	Orlop stringer plate S F A Three D A+B P+I P stringers.		
"	28½	"	29½	20½ "	21½	....		"	"	"	10 E 12 E	P S+S F A P S+D F A 4"×3½"+F P	....	....
Steam.	19½	"	22	....		....		"	Alt. 8 E 10 E	Plate P S+S F A P S+D F A 3½"×3½"+F P	....	...	....	....
"	22	"	23	....		....		"	Alt. 8 E 10 E	Plate P S+S F A P S+D F A 3½"×3½"+F P	....	Double angle.	....	....
"	23	"	24	....		....		"	Alt. 8 E 10 E	Plate P S+S F A P S+D F A 3½"×3½"+F P	....	D A+B P	....	....
"	24	"	26	17 "	18	....		"	Alt.	Plate.	10 E 2+4 Alt. 8 E 10 E	P S+S F A Plate P S+S F A P S+D F A 3½"×3½"+F P	....	....
"	26	"	27	18 "	19	....		"	"	"	2+4 Alt. 8 E 10 E	Plate P S+S F A P S+D F A 4"×3½"+F P	....	....
"	27	"	28	19 "	20	....		"	"	"	Alt. 8 E 10 E	Plate P S+S P A P S+D F A 4"×4"+F P	....	....
"	28	"	30	20 "	22	....		"	"	"	Alt. 8 E 10 E	Plate P S+S F A P S+D F A 4"×4"+F P	....	D A+B P
"	30	"	32½	22 "	24½	....		"	"	"	Alt.	Plate.	....	P S+S F A
"	32½	"	36	....		17½ under 21		"	"	"	"	"	10 E	P S+D F A
"	36	"	39	....		21 up.		"	"	"				

Single face bars to equal keelson bars. Bulb plates to equal depth of deck beam.
* To top of main-deck beams in awning-decked vessels of less than 7½ feet in depth.
D A = double angle; B P = bulb plate; I P = intercostal plate; D F A = double face angle; P S = plate stringer; S E A = single face angle; F P = face plate; E = extra heavy.
Alt = alternate frames: 2+4 = second and fourth frames alternately: 8E = on every eighth frame, if extra heavy beams are used: plus sign (+) means *with*.
Example: P S + D F A 3½" x 3½" + F P means: plate stringer with 3½" + 3½" double face angle and a face plate.

17. Beam Knees.

1. Welded or plate beam knees to be not less than three beam depths, as follows:

a. Upper deck beams on steamers having only one tier of beams.

b. Middle deck beams on steamers having deep frames or web frames in lieu of a tier of beams.

c. When beams are supported by three rows of pillars.

d. Upper deck beams in way of omitted middle deck half beams.

e. Deck beams in water-tight flats and peak tanks.

f. Upper decks, lower decks, orlop deck beams of sailing ship.

2. When two rows of beam pillars are fitted, 2 3-4 depth of beam. In all other cases, 2 1-2 depth of beam Knees to be not less than 6-10 of depth across throat.

TABLE 9.—RIVETING OF BEAM KNEES.

Depth of Knees.	Rivets. No.	Rivets. Dia.
Under 17 inches	4	¾
17 inches and " 21 "	5	¾
21 " " " 24 "	5	⅞
24 " " " 28 "	6	⅞
28 " " " 32 "	7	⅞
32 " " " 36 "	8	⅞
36 " " " 40 "	9	⅞

18. Deck Plating, Stringer, and Tie Plates.

1. Upper decks to be plated in way of hatches 22 feet or more in length, with plating tapering to deck stringer. When either the engine or boiler opening exceeds 15 feet or combined length of both exceeds 30 feet, the beams in way of same should be plated over, the plating extending two beam spaces beyond the openings and tapering thence to stringer plate for a distance equal to width of plating required to be fitted. When large openings are adjacent the deck space between should be plated.

2. When a full or half deck is required, it is to be fitted to upper decks in one and two-deck vessels and to the middle or upper deck in three and spar-deck vessels. (Steel decks should be increased in weight at all deck openings, with doubling plates at corners of large deck openings.)

3. When a half steel deck is laid it is to maintain the full breadth of ship for 1-2 length amidships and thence taper at each end for 1-8 of vessel's length into deck stringer.

4. When only one steel deck is called for, the stringer plate may be reduced in width one inch for every 7 feet of ship's length. Orlop stringer plates to be 3-4 width of lower hold beam stringer plate.

5. Stringer plates to maintain full midship size for 1-2 length amidships.

6. Fore-aft tie plates to be worked on each beam tier at side of hatches. When hatch ways are 16 to 20 feet in length the width of said plates to be double that for hatch of ordinary width, extending for a length of two beam spaces beyond each end of hatch.

a. Diagonal tie plates at mast partners of sailing vessels to be of same weight as corresponding deck stringer; and in said vessels, whose second number is or above 15,000, to be fitted on upper deck throughout.

7. Steel decks should be fitted on bulb beams when fitted to alternate frames in vessels of over 34 feet beam.

8. MAST PARTNERS.—Plates to equal deck stringer in weight and be not less than 3 mast dias. in width. Rings for sailing vessels to be 1 inch deeper than required for bulb angle frames. For steamer rings to equal frame angle.

19. Stanchions.

TABLE 10.—ROWS OF STANCHIONS AND NUMBER OF RIVETS IN EACH END.

Beam Length Amidships.	Rows of Stanchions.	Stanchions. No. and Size Rivets.	
		Length or Dias.	
Over 43′	2	10′ under 18′ 2⅝″ under 4″	Two ⅞″
Over 55′	3	18′ " 24′ 4″ or over.	" 1″
....	....	over 24′	Three 1″

1. When beams are spaced on every frame, stanchions are to be fitted at alternate beams, with a double angle strong back, equal to reverse bars, worked under said beams and clipped to same.

2. In deep tank, with center line bulkhead, a row of quarter stanchions attached to an intercostal girder is to be used.

3. Additional stanchions to be placed under deck houses, heel of bowsprit, windlass, steam winches, and capstans.

4. Bulwark stanchions to be spaced not more than 6 feet apart, and in sailing vessels of 1,800 tons or over not more than 4 to 5 feet apart. To be from 1 3-8 inches to 2 inches diameter, with four 7-8-inch rivets or taps at heel. To have spur to bulwark with doubling plate at same.

20. Bulkheads.

1. Forward collision bulkhead worked to upper, spar, or awning deck to be at least 1-20 vessel's length aft of stem. Aft collision bulkhead to be at a reasonable distance forward of stern. A water-tight bulkhead shall be fitted at each end of machinery space.

TABLE 11.—NUMBER OF W. T. BULKHEADS IN CARGO SPACES AND TOTAL NUMBER IN SHIP.

Length of Ship.	220′* under 280′	280′ under 330′	330′ under 400′	400′ under 470′	470′ under 540′	540′ under 600′	600′ under
Forward cargo space.....	1	1	1	2	2	3	
Aft cargo space.....	0	0	1	1	2	2	
Total in ship.....	4	5	6	7	8	9	

* Machinery placed aft.

2. Said cargo and machinery space bulkheads to be carried up to upper decks of vessels having one, two, or three decks; to spar deck in spar-deck vessels, and to main deck in awning-deck vessels. The latter having deep web frames worked between decks over bulkheads below.

3. All frames on water-tight bulkheads shall be doubled. All angles connecting bulkheads to decks or double bottom plating are to be doubled and equal to reverse frames.

4. Stiffeners on hold bulkhead to be of angle bars equal to size of main frames, worked vertically on one side, not more than 2 feet 6 inches apart, and horizontally on the other not more than 4 feet apart. Vertical stiffeners to be well secured to floor plates, or where double bottoms are used to be bracketed to same. When bulb angles are used for horizontal stiffeners they are to be bracketed to shell.

5. Bulkhead shell liners are to be wide enough to take the frames forward and aft of bulkhead frame, or they may be of approved diamond shape.

6. All collision bulkheads, and all other bulkheads 40 feet or above in breadth shall have their horizontal stiffeners of bulb angles 1 inch deeper than specified for bulb frames. They shall be bracketed to shell. All bulkheads of 36 feet and under 45 feet in width to have in addition a vertical web plate at center, from center keelson to lower deck beams. When of 45 feet and under 55 feet in width to have two such web plate stiffeners. When of 55 feet and over 60 feet in width to have three such web plate stiffeners.

7. Sailing ships need have but the forward collision bulkhead.

8. Center line bulkheads shall have vertical stiffeners on each side two frame spaces apart. Those on one side being attached to beams. They are to have double top and bottom angles of at least 3 inches by 3 inches by 7-16 inch.

9. Bulkheads in vessels requiring lower deck, hold, or orlop beams, which are not supported on both sides by a lower or orlop deck, to have a semi-box girder horizontal stiffener. When web frames are used in lieu of hold beams all transverse bulkheads are to have a center vertical web and semi-box beam in addition to regular stiffeners.

10. Bulkheads at fore end of all bridge houses and of poop extending over engine and boiler or other deck openings to have a coaming—plate higher and 2 pounds heavier than plating. Plating to equal that of bridge house side plating and be stiffened with bulb beams 1 inch deeper than required for bulb angle frames, 30 inches apart, bracketed at each end to decks. If said poop or bridge house does not extend over a boiler, engine, or other opening the stiffener may be of angle bars 30 inches apart.

11. Bulkheads at front of brake of a raised quarter deck are to have an athwart ship stiffener plate, equal to upper deck beam tie plates, riveted at ends to deck ends with brackets to bulkhead's stiffening angles. Said bulkhead plating to equal that of bridge house side plating and have angle stiffeners equal to frames, 30 inches apart.

21. Shaft Tunnel.

Plating to equal that of lower half of bulkheads. Stiffeners to be spaced not more than 4 feet apart (3 feet under hatches), and equal to reverse bars. Stiffeners on recess at aft end to be spaced as frames. Top of tunnel under hatches to be 2-20 inch thicker than rest or have a 2-inch wood covering. Water-tight doors at forward end, to work from upper deck. When stanchions to beams are located on shaft tunnel, the tunnel must have additional stiffening at said point.

22. Breast Hooks.

1. Side stringers in end peaks should be spaced about 4 feet apart. When second number is or over 24,000 breast hooks to be fitted between each deck tier, at both ends. Depth for regulating the tier of alternate frame deck or panting beams in fore peaks to be taken at forward collision bulkhead. Rules under Section 16 to regulate said beams.

2. Sailing Vessels under 20 feet depth at collision bulkhead to have lower deck or panting beams at alternate frames. When said depth is 20 feet and under 30 1-2 feet, to have additional tier of panting beams below lower deck tier. When said depth is 30 1-2 feet and under 33 feet to have two tier of panting beams below lower deck tier. Stringer plates on panting beams should extend aft of collision bulkhead 1-4 of vessel's amidship beam and be in width 2-3 of midship width of lower deck stringer.

23. Shell Plating.

1. Plates should not be less than 6 frame spaces in length except at ends. To maintain midships weight for 1-2 length amidships. Forward and aft of said 1-2 length the plating may be gradually reduced by 1-20 of an inch to thickness specified for plating at extreme end. In sailing vessels, the overlapping strakes for 1-4 length at forward end are to be only 1-20 of an inch less in thickness than midship plates.

2. Butt of adjoining strakes to be at least 2 frame spaces apart; of alternate strakes 1 frame space. Butts of garboards to be shifted clean of keel scarphs and of each other at least 2 frame spaces. Butt of deck stringers should be at least 2 frame spaces from butts of their sheer strakes.

3. Sheer strakes of one, two, three, and of spar-deck vessels to extend above gunwale bar to allow of two horizontal rows of rivets in butt straps above said bar.

4. (*a*) Boss plates are to be at least the same weight as the midship plates of the same strakes. When second number is 13,900 and under 26,500 said plates to be 1-20th heavier than midship plates, and have treble-riveted butts if said number is 18,700 and under 26,500. When said number is 26,500 and over, said boss plate and the one below and above to be 2-20-inch heavier than the midship plates, with treble-riveted double straps, or boss plates are to be worked double. (*b*) Garboard plates to stern post in screw vessels to be of the same weight as those amidship. In vessels whose second number is 16,600 or above, all plates connecting to stern post to have same weight as their respective strakes amidships.

5. In ships of full form the two strakes next to garboards to maintain full midship weight up to collision bulkhead.

6. Sailing vessels to have three bilge strakes increased 1-20 inch, all fore-aft, when second number is 16,000 and above. When said number is 22,000 or above the strakes in way of hold beams are to be also increased 1-20 inch for 1-2 length amidships.

7. When scuttles are fitted in sheer strake within 3-5 length amidships additional strength must be provided at said point.

24. Butt Straps.

1. Single butt straps to at least equal the thickness of plates they connect.
   Double butt straps to be 3-20 inch thicker than 1-2 thickness of plates they connect.
2. Butt angle straps to keelsons bars must be at least 2 feet long.
3. Butt straps of all vertical keelson plates, floors, and web frames must be worked double.
4. Butt straps of flat keel plates to be as much thicker than keel plates as is required for bilge strakes.
5. Butt straps for masts to be 1-20 inch thicker for double riveting and 2-20 inch for triple riveting than plates they connect. They should be fitted on outside of masts and bowsprits. Straps of topmasts to be 1-20 inch thicker than plates they connect.
6. Shell and Deck Stringers.. Also see Table 13.

a. When outside strake plates are above 40 inches and not over 46 inches in width, or the inside strake plates are 48 inches and not over 54 inches in width, their straps are to be 1-20 inch thicker than plates they connect, unless they are specified to be 4-20 inch, as stated in table 13.

(To be continued.)

New Tug Boat Cornell.

The latest addition to the fleet of the Cornell Steamboat Company is the steel tug boat *Cornell,* recently completed at the yard of the Townsend-Downey Company from designs prepared by Mr. J. A. Hargan. The dimensions of the vessel are not given, but suffice it to say that she is of the largest type of sea-going tug boat. The hull is built of steel throughout and subdivided into five water-tight compartments by four thwartship bulkheads. The frames are spaced 22 inches, center to center, and are of angles 3 1-2 by 3 inches by 7 pounds. The deck beams are of channel bars 6 by 2.81 by 2.81 inches by 13.3 pounds and are placed at every frame. The stem and stern posts are of the best hammered iron, the former 7 by 2 1-4 inches in section, and the latter 7 1-2 by 2 3-4 inches. The flat plate keel is 44 inches wide by 18 1-2 pounds weight per foot. The bilge plating is 17 1-2 pounds, the sides and bottom are laid with 15-pound plates, and the shear strake is 18 1-2 pounds. The deck is laid with yellow pine, 3 1-2 by 3 1-2 inches.

The compound condensing engine was built by the Cornell Steamboat Company at the Rondout shops. The high-pressure cylinder is 20 inches diameter and fitted with a piston valve; the low-pressure cylinder, 53 inches in diameter, has a flat, double-ported slide valve. The cut-off in the high-pressure is 22 1-2 inches and in the low-pressure 22 inches. The common stroke is 36 inches. A complete set of Blake pumps is installed, and the air pump is independent, of the same make. The propeller is of cast iron, 10 feet 3 inches diameter by 15 feet pitch, with 4 blades of maximum width, 56 inches. When towing, the engines making 97 revolutions, indicate about 1,390 horse power; while when running light, at 135 revolutions, the horse power is 1,934, and the speed in the neighborhood of 18 miles. The total fuel capacity is 190 tons, and it is stated that 3,480 pounds of buckwheat anthracite are burned per hour when the vessel is towing, and 4,835 pounds when running light.

Steam is supplied by two single-end Scotch boilers, built by Richard Hammond, of Buffalo, N. Y., 12 feet 1 3-8 inches long by 14 feet 9 inches mean diameter, for a working pressure of 180 pounds. The average pressure carried at the boilers is 175 pounds, which is reduced 5 pounds at the engines. Each boiler is fitted with three Morison suspension furnaces, 46 inches inside diameter. The donkey boiler, located in the deck house, midship, is the vertical tubular type, 36 inches diameter by 5 feet 10 inches high.

The vessel is lighted throughout by electricity generated by a direct-connected, 7 1-2 Kw. set, of the General Electric make. The deck machinery includes an American ship windlass and capstan, and a Williamson steam and hand-steering gear.

OBITUARY.

Mr. John Edwin Thropp died at his home in Trenton, N. J., on November 24, 1904. Mr. Thropp was prominently identified in the trade of marine engine builders for many years, and established the firm of John E. Thropp Sons Company, of which he was president at the time of his death.

Mr. James H. Williams died on December 8, at his home, 6 Pierrepont street, Brooklyn. Mr. Williams was born in Fort Plain, N. Y., fifty-nine years ago. He was prominently connected with James H. Williams & Company, of Brooklyn, manufacturers of drop forgings, which he established in 1883.

Report of Boiler Makers and Iron Shipbuilders.—The report of the boiler makers and iron shipbuilders is disheartening, from the state of trade point of view, says *London Engineering.* There is very little as yet to indicate any improvement in trade, but the report states that there is a more hopeful feeling in some districts, and it is thought that a revival is near at hand. The quarterly report shows a total membership of 48,804. The union has been supporting a vast army of sick, unemployed, and superannuated members to the extent of 9,964.

QUERIES AND ANSWERS.

All reasonable questions concerning marine engineering received from and signed by subscribers will be answered by the editor in this column. All communications must bear the name and address of the writer.

Q. 258.—Please answer in your next issue the following questions: I have a launch of the torpedo type, 30 feet long, 6 feet beam, with a draft light—12 inches—which I built myself. It is very light, bow rather fine lines; of course the propeller hangs a little below the keel. I also have an engine of two cylinders, of the four-cycle type, bore 5 1-4 by 6 inches stroke, which, when running at 900 revolutions, gives a brake horse power of 12. The hull alone weighs 1,800 pounds. Can I run a propeller at this speed with good results? If so what size of pitch and number of blades shall I use and at what speed should I expect the boat to run?

A.—In lack of information, we assume that your engine will weigh 500 pounds; that your gasoline stores and equipment will weigh 200; and that the two persons on board will weigh 300, making a total weight of hull, machinery, and load, of 2,800 pounds. With a well-designed, lightly-built hull, a high-speed engine of 12 horse power should be able to drive this boat at a speed of about 12 1-2 miles per hour.

There is no difficulty in designing a propeller to run at 900 revolutions—in fact, higher speeds than this are now being considered*—but we, however, would not recommend very much over 1,000, owing to cavitation, which sets in at excessive speeds. With the power and revolutions stated, a four-bladed wheel of about 16 inches diameter and 19 inches pitch should prove to be about right. The blades should be elliptical and their combined developed area should be about 38 per cent., the area of a disk of the same diameter. An engine designed for 900 revolutions would probably not give satisfaction when running at much greater speeds, as there will be too excessive wire drawing of the gases passing through the valves, and you may expect difficulty with your carburetor and igniter. There may be structural reasons why your engine should not be run at an excessive speed, and in regard to this point the manufacturer should be consulted.

Q. 267.—Please tell me in the Q. & A. column what I would gain by coiling my feed pipe around in the steam space of the boiler, before it discharges into the water. W. D. H.

A.—The advantage to be gained by coiling your feed pipe in the steam space of boiler would be that the feed would be brought to a high temperature before discharging into the boiler, providing the coil was sufficiently long. This would mean that there would be less internal strain upon the boiler due to unequal temperatures of cold feed. This system alone is hardly sufficient, as there is no provision made for removing the mud or scale-forming deposit which generally comes with heating the water. Another objection is that the feed can in no way be heated until after steam is raised, and cannot be of service, such as the hydrokineter, for circulating the water in the boiler while raising steam. These remarks apply alone to large boilers, for such requirements are scarcely considered necessary in small boilers. We would refer you to two articles on feed water heaters which recently appeared in MARINE ENGINEERING, one on the internal coil type, page 386, July, 1903; another on somewhat similar design, page 90, February, 1904.

Q. 268.—Please answer the following question through MARINE ENGINEERING and oblige. What amount of coal, in tons, does the *Kaiser Wilhelm II.* consume per day when crossing the Atlantic, *i.e.,* approximately, her consumption for 24 hours? R. K. B.

A.—The indicated horse power of the main engines of the *Kaiser Wilhelm II.* is about 38,000. Assuming that for all purposes 1 3-4 pounds of coal per I. H. P. of main engines are burned, this means that about 30 tons per hour, or 720 tons per day are shoveled into the furnaces.

*See paper of Lieut. W. D. Taylor, presented before the November meeting of the Society of Naval Architects and Marine Engineers. The discussion appeared in the December, 1904, issue of MARINE ENGINEERING.

Q. 269.—Will you please answer in your Q. & A. column the following question: Suppose we have a compound engine 12 by 24 by 20 inches, boiler pressure 110, and a pressure of 4 pounds on the low-pressure cylinder, having no cut-off on the low-pressure cylinder, we will put one on and cut it off till we have a pressure of 16 pounds on the low-pressure cylinder. Am I gaining anything? I cannot see as I am. I should think that it would make more of a back pressure for the high to carry. W. M. F.

A.—If by "4 pounds on the low-pressure" you mean an exhaust pressure of 4 pounds, your calculations are correct. The higher the back pressure the more you will cut down the power of the engine. You can readily see this by assuming a still more exaggerated case than yours—for instance, that the back pressure will be 75 pounds. The effective pressure of the engine is determined, among other conditions, by the range of pressures—that is, the difference between the admission and the exhaust pressures. In your case it is 110 — 16 = 94 pounds. As a matter of fact, 1 pound pressure at the lower end of the expansion is more effective than it is at the initial end. For instance, if you reduce your back pressure from 4 to 3 pounds, you have an added effective pressure of 1 pound working in your low cylinder. As the area of the low is four times the area of the high, this pound of pressure will give, relatively speaking, four times the amount of work as 1 pound increased initial pressure in the high-pressure cylinder. This is not strictly correct, but serves for illustrating the point.

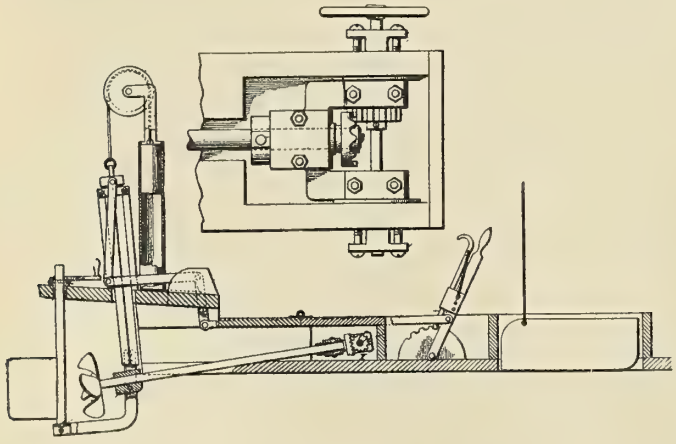
Ferryboats.—Ferryboats which ply between Oakland and San Francisco are the largest of this type of boat in the United States. The first of these large boats was built in 1898. It is propelled by triple-expansion engines, is 284 feet long, has a 64-foot beam, 17 feet depth of hold, and 1,945 gross tons of displacement.

SELECTED MARINE PATENTS.

772,383. WATER MOTOR. HERMANN J. M. SIEMERS, HAMBURG-HORN, GERMANY.

Claim.—1. In a water motor, the combination of a body provided with air chambers, with a water-inlet valve, and a suction pipe having a perforated lower end and adapted to close said valve. Three claims.

772,384. BOAT AND PROPELLING MEANS THEREFOR. FREDERICK W. SMITH, ELLSWORTH, MICH.

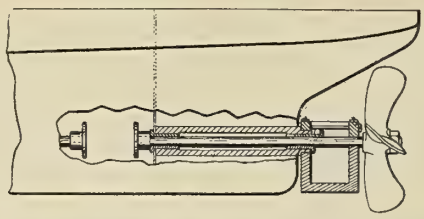


Claim.—1. The combination, in a boat, of a shallow body having an elevated deck on its rear portion, a hanger movable up and down through the elevated deck of the body, suitable means for adjustably fixing the said hanger, a propeller carried by and movable with the hanger, and means connected with the propeller for transmitting rotary motion to the same.

772,970. BUOYANCY-REGULATING APPARATUS FOR SUBMARINE BOATS. LAWRENCE Y. SPEAR, GREENPORT, N. Y., ASSIGNOR TO ELECTRIC BOAT COMPANY, NEW YORK, N. Y.

Claim.—1. A submarine boat having means for regulating and maintaining the reserve or surplus buoyancy of the boat, said means comprising a plurality of connected adjusting tanks of known capacity, means for filling one or more of said tanks from water of flotation, means for blowing out the water from one or more of said tanks, and gauges, actuated by a hydrometer, showing the weight required to put the boat in diving trim and the changes in displacement due to differences in the specific gravity of the water of flotation.

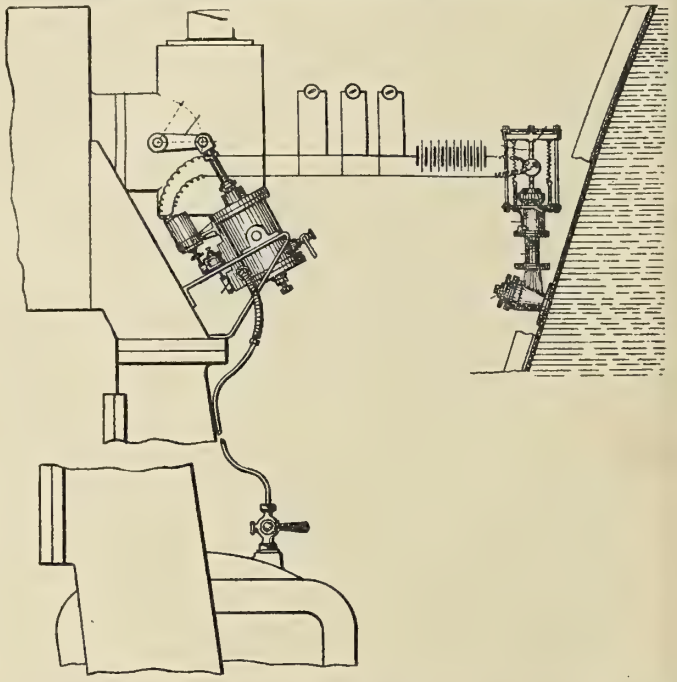
773,036. APPARATUS FOR RENEWING THE STERN BUSHINGS OF SHIP PROPELLER SHAFTS. HERMAN SMITH, LUDINGTON, MICHIGAN.



Claim.—1. The combination with a ship of a removable receptacle open at the top and located between the propeller wheel and the stern wall of the ship, with the propeller shaft extending through suitable apertures in the wall of the receptacle and with the shaft tube fitting and projecting into the aperture of the inner receptacle wall.

773,079. MARINE-ENGINE GOVERNOR. ANDREW KERR, NEW BRIGHTON, ENGLAND.

Claim.—1. In an automatic governor for marine engines, the combination of a cylinder arranged below the water-level in the tunnel of a ship, near the propeller, and having a plurality of water inlets connected therewith, a cock for controlling the escape of air from the cylinder, a water-tight piston within the cylinder having a plurality of arms mounted

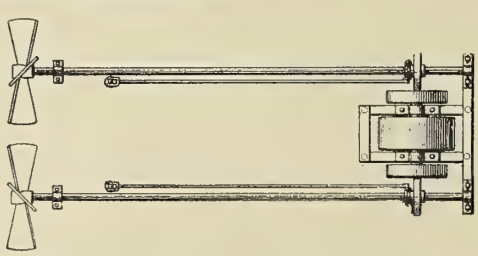


on its rod outside of said cylinder, a bar connected to and projecting on opposite sides of said piston rod and engaging suitable stationary guides, springs adjustably connected to said bar on opposite sides of the piston-rod and adapted to regulate the stroke of the piston, an electric switch having pawls projecting into the path of the arms on the piston-rod, means for controlling the supply of steam to the engine, a source of electricity, and an electric circuit including said source of electricity, said switch, and said means for supplying steam to the engine.

773,146. SUBMARINE VESSEL. WILLIAM HUGUET, LOUIS MINART AND FRANCOIS MIRON, PARIS, FRANCE.

Claim.—1. A submarine apparatus provided with external tools and electric motors, and having a perfectly-water-tight shell without any moving parts passing through it, whereby the external tools may be operated under water at great depths while the internal pressure is only that of the atmosphere.

773,615. FRICTIONAL GEARING FOR MOTORS. EDWARD W. WICKEY, EAST CHICAGO, IND.



Claim.—1. The combination with a driving shaft and opposite frictional disks perpendicular thereto, of oppositely disposed frictional pinions perpendicular to and bearing upon said disks, whereby the pressure of one pinion balances that of the other, and means for moving said pinions radially upon the disks independently of each other and across the center of the frictional surfaces.

773,285. DEVICE FOR UTILIZING THE POWER OF UNDULATIONS OF WAVES OF THE SEA. FRITZ GOEDECKE, TELTOW, NEAR BERLIN, GERMANY.

Claim.—1. The combination with a screw spindle, of a float guided to move vertically, a clutch, and a sleeve on the spindle adapted to be engaged by the clutch and drive the spindle, substantially as and for the purposes set forth.